(THIS IS A REPRINT, WITHOUT CHANGE, OF ORDP 20-347, WHICH WAS REDESIGNATED AMCP 706-347.)

ENGINEERING DESIGN

HANDBOOK

CARRIAGES AND MOUNTS SERIES

TRAVERSING MECHANISMS

TABLE OF CONTENTS

TABLE OF CONTENTS (Concluded)

LIST OF ILLUSTRATIONS

LIST OF TABLES

LIST OF SYMBOLS

A_b effective area of buffer piston

A_c surface area of clutch

A_s surface area of brake shoe

a_o orifice area

C_F ratio, face width to circular pitch

C_i gear tooth inbuilt factor

CG center of gravity

c_o orifice coefficient

c_r Barth's velocity factor

D brake diameter

D_b effective diameter of thrust bearing

D_p pitch diameter of gear

D_w pitch diameter of worm

d_g depth of orifice groove

E_a energy absorption rate

E_b energy absorbed by brake

E_r energy rate

E_t maximum kinetic energy of traversing parts

F_a inertia force of recoiling parts

F_b buffer force

F_{ba} applied brake force

F_{br} normal load on traversing bearing

F_{ca} applied clutch force

F_e effective face width of gear

F_g propellant gas force; gear tooth load

F_{gs} limiting gear tooth load for strength

F_{gw} limiting gear tooth load for wear

F_N force normal to surface

F_s spring load

F_w face width of gear tooth

F_{xx} gear tooth load, general expression

HP horsepower

K recoil force

K_w wear factor of gear tooth

L subscript denoting left brake shoe

M mass of gear

M_f brake shoe frictional moment

M_F applied brake shoe moment

M_N brake shoe pressure resisting moment

M_w weight moment, traverse

N number of gear teeth

N_w number of threads in worm

n number of gear meshes in train

P_c circular pitch

P_d diametral pitch

P_L axial pitch of worm

P_m power generated by motor

P_t power required to rotate traversing parts

p_c clutch contact pressure

p_m maximum pressure on brake band

R subscript denoting right brake shoe

R_b radius, traversing axis to line of action of buffer

R_{p_x} pitch radius of gear, general expression

R_t radius, traversing axis to CG

R_x radius, buffer contact point to traversing axis

R_μ radius of friction circle

r_g gear train ratio

r_i inside radius of bearing

r_u ratio of peak speeds, motor to mount

r_o outside radius of bearing

S_b total buffer stroke

s endurance limit in bending

T torque output required of power source

T_a accelerating torque of traversing parts

T_b frictional torque of traversing bearing

T_{bs} braking torque per shoe

T_c torque transmitted by clutch

T_f component of firing torque in plane of traverse

T_R torque of traversing gear

T_s applied torque of antibacklash device

T_t torque at power source converted from that of traversing parts

T_w torque on worm gear

T_{xx} torque on gear unit, general expression

T'_{xx} accelerating torque of each gear unit, general expression

T_α accelerating torque of gear train

t_b braking time

t_m time constant of motor

t_t time constant of traversing parts

v_x linear velocity of buffer

W_{tr} weight of traversing parts

w brake band width; density of hydraulic fluid

x subscript denoting gear number, general expression

LIST OF SYMBOLS (Concluded)

x_b buffer stroke at any position θ_b

xx subscript denoting gear unit, general expression

y gear tooth form factor

y' gear tooth form factor, dynamic load

α maximum traversing acceleration

α_b deceleration of traversing parts induced by buffers

α_{br} deceleration at brake

α_{tb} braking deceleration of traversing parts

α_{xx} acceleration of gear unit, general expression

β gear pressure angle

γ ratio of motor speeds, maximum operating to peak

Δ deflection of spring

δ density of steel

η gear train efficiency

η_g efficiency of each spur or bevel gear mesh

η_w efficiency of worm gear

θ angle of elevation

θ_b angular buffing distance

θ_{bm} maximum angular buffing distance

θ_{br} brake drum travel

θ_t slope of terrain

λ lead angle of worm gear; helix angle

μ coefficient of friction, general expression

μ_b coefficient of friction of thrust bearing

ρ mass density of hydraulic fluid

σ general expression for stress

σ_c endurance limit

ϕ_a position of CG of traversing parts relative to horizontal

Φ mass moment of inertia of traversing parts

Φ_c effective mass moment of gear

Φ_{xx} mass moment of inertia of gear unit, general expression

ω maximum angular velocity of traversing parts

ω_{xx} angular velocity of gear, general expression

ω_b angular velocity of traversing parts during buffing

ω_{br} angular velocity of brake

CARRIAGES AND MOUNTS SERIES
TRAVERSING MECHANISMS*

I. INTRODUCTION

A. PURPOSE

1. This is one of a series of handbooks on carriages and mounts. This handbook covers traversing mechanisms, one type of which is shown in Figure 1. The various types of mechanism are discussed, along with their operating characteristics and pertinent design data. Contents is devoted exclusively to traversing mechanisms except where limited descriptions of other elements of carriages and mounts are necessary to clarify the discussions. More elaborate descriptions of these elements appear in other Ordnance Corps Pamphlets of the Carriages and Mounts Series (ORDP 20-340 through ORDP 20-348). Since traversing and elevating mechanisms have much the same requirements and characteristics and use similar mechanisms, some of the material contained in this handbook is duplicated in ORDP 20-346, *Elevating Mechanisms*, to avoid excessive cross-reference.

B. GENERAL DISCUSSION

2. The long range of projectiles and missiles requires extremely accurate aiming. Slight errors in the pointing of either cannon or launcher ramp can result in considerable deviation at the target. When aimed by direct sighting, a weapon must be moved slowly and precisely to align it accurately with the target. When aimed by a separate fire control unit, the cannon or the missile launcher must be able to respond accurately to the direction signals of the unit. In either case, such a weapon is too heavy to be pushed by hand into the proper horizontal direction. Hence, a manual or

power operated mechanism is provided to enable a gunner to attain a precise position and to hold it there during firing. On the other hand, many small arms are light enough to be moved into position without the aid of a mechanism.

3. Position coordinates for the gun tube are given in angles about two axes. Normally one coordinate is in elevation about a horizontal axis and the other is in traverse about a vertical axis.

4. The components of a weapon which traverse are called the traversing parts and generally consist of the cannon and other tipping parts and the top carriage. The unit which rotates the traversing parts, which controls this rotation, and which holds the cannon firmly in azimuth during firing is the traversing mechanism. It and its controls must be designed for easy operation to the extent that the gunner can devote most of his attention to maintaining proper weapon orientation. An optimum traversing mechanism must combine precision, reliability, durability, speed, and low-power demand, with ease of control. For some installations, it must include high acceleration, yet must be able to stop smoothly without overrun.

5. Essential mechanical characteristics of traversing mechanisms on missile launchers are similar to those on guns. Although they are generally subjected to less severe loads, they must be shielded from the rocket blast or construction and materials used must be capable of withstanding the blast.

6. Some missile launchers on airplanes are similar to bomb launchers. These have no elevating or traversing mechanism, aiming being done by positioning the entire airplane. Such launchers can be built to carry either
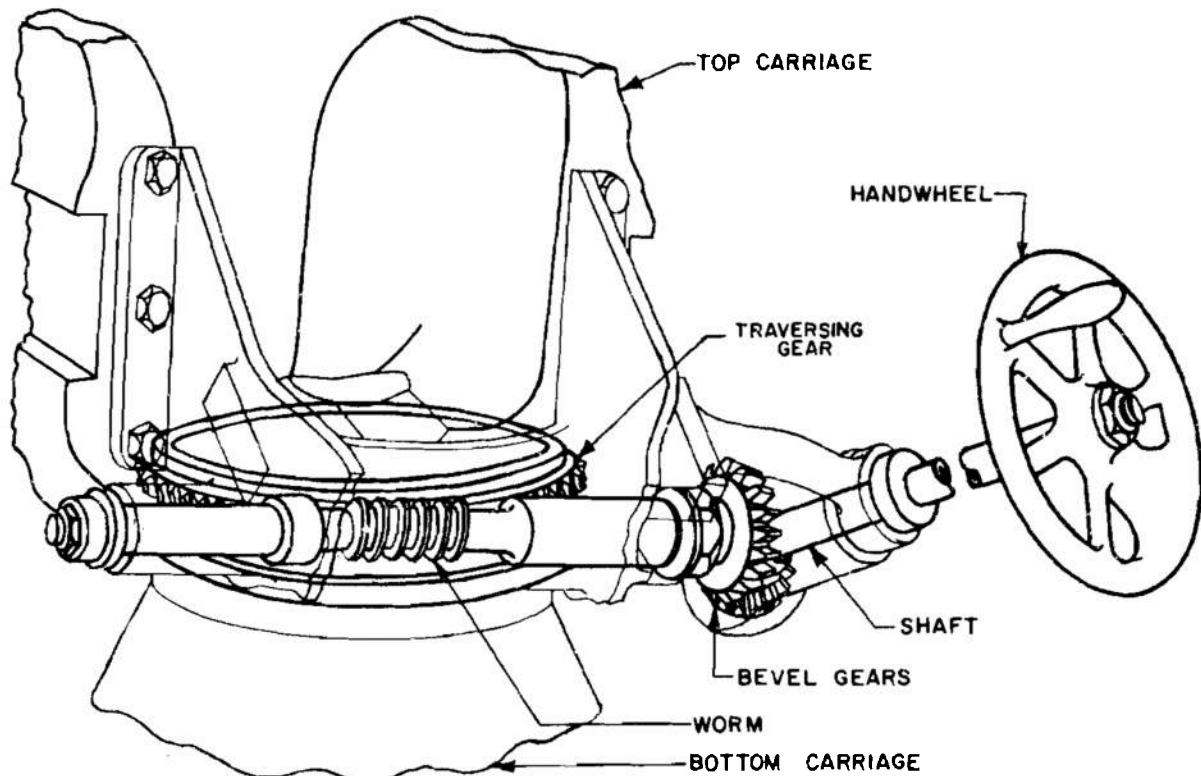
Figure 1. Manually Operated Traversing Mechanism

bombs or rockets. Others are intended for air-to-air combat where a flight path parallel to that of the target will permit closer approach than one coincident with the target path. For such use it is necessary to provide for traversing the launcher to intercept the target.

7. Launchers for ground-to-air or ship-to-air missiles are required to traverse and elevate at high speeds to pick up, track, and intercept air targets. The requirements for their structures and mechanisms are equivalent to those of anti-aircraft guns; in fact, it has been practical to convert some gun mounts into missile launchers. Ship-mounted antiaircraft guns and missile launchers place especially high burdens on traversing and elevating mechanisms. In addition to tracking high-speed air targets, the two mechanisms must provide continuous compensation for motion of the firing platform, that is for rolling, pitching and yawing of the ship. The latter function is frequently more critical and demands higher rotational velocities than does target tracking.

II. TYPES OF TRAVERSING MECHANISM

8. Because of the close relationship between the forms of traverse mechanisms and the type of traverse bearing, the mechanisms are classified according to this relationship, namely:

> A. Axle
> B. Pintle
> C. Base Ring and Racer
> D. Ball Joint

Another type, now obsolete, is the railway gun that was traversed by moving it along a curved track.

A. AXLE TRAVERSE

9. The axle traverse mechanism is used on a single trail mount. Traversing parts slide along the axle. Some weapons use rollers to transmit the carriage weight to the axle thus minimizing frictional resistance between sliding members. Coarse traverse is achieved simply by lifting the trail and pivoting the

weapon on its wheels until aimed in the general direction. During this activity, the traversing mechanism locks the top carriage to the axle. With the wheels on the ground and the axle fixed laterally, fine aiming is achieved by moving the carriage along the axle with the trail spade serving as the pivot. Some mounts have firing jacks to take the load off the wheels after emplacement. Since the weight of the cannon, recoil mechanism, carriage, and part of the trails is supported on the axle, rollers may be provided to permit the lateral motion along the axle with low friction to facilitate traversing.

10. Fine traverse or accurate laying of the gun is the function of the traversing mechanism which here utilizes the principle of the screw and nut. The screw thread may be on the axle with the nut captive on the carriage or the nut may be fixed on the axle with the screw captive on the carriage (see Figure 2). In either case, rotation of nut or screw is accomplished through shafts, gearing, and universal joints from a handwheel located on the carriage. The mechanism on the 75 mm Pack Howitzer, M1A1, utilizes a ball bearing screw to minimize friction.

11. By convention, the handwheel is located on the left side of the carriage. It must be in a convenient position for the gunner and must require little effort to operate while sighting. Minimum effort is provided by the mechanical advantage of the gear train. Because of the short axle and the interference of the wheels, the traversing range is restricted to about ±3 degrees.

B. PINTLE TRAVERSE

1. Description

12. A more versatile mechanism adaptable to any type of mount and providing a greater range of traverse is one in which the traversing parts pivot on the bottom carriage, axle, or base, about a pintle (see Figure 3). The pintle is attached to the top carriage and rotates in a bearing on the bottom carriage. The traversing parts are supported by a thrust bearing. Where clearances permit, 360° traverse is available. However, the stability of the weapon usually determines this range. Weapons which are stable in all directions may have unlimited traverse while those, such as the split trail type, which are stable only within the spread of the trails are limited to about 90° traverse (see Figure 4).
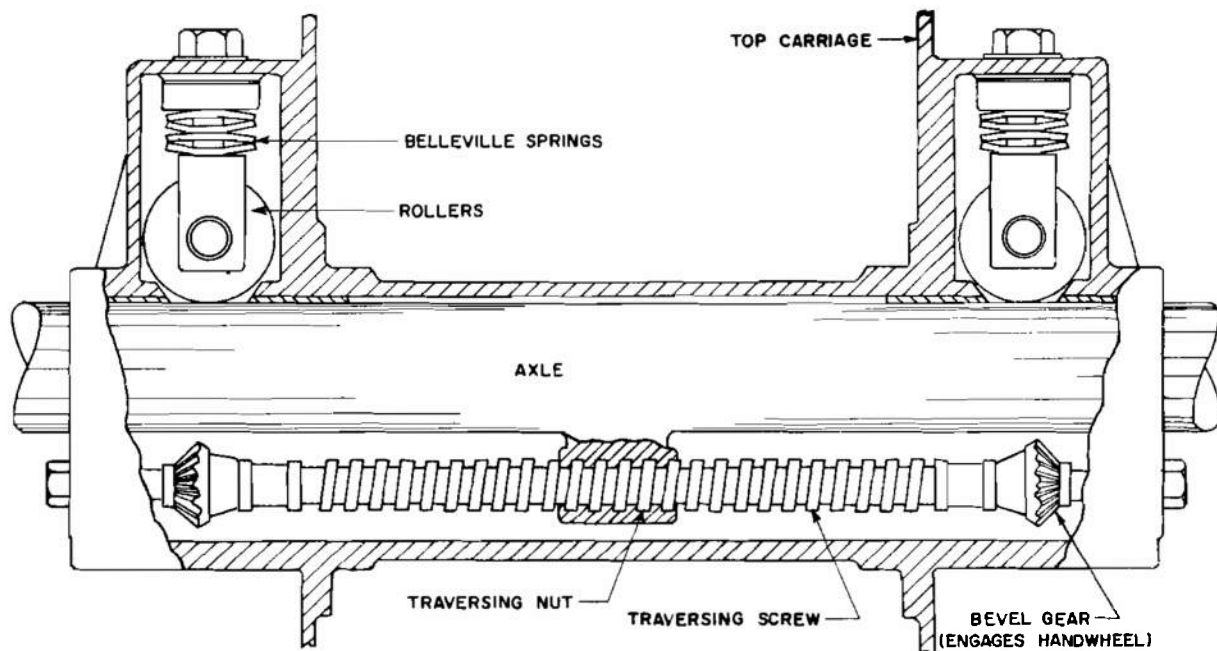

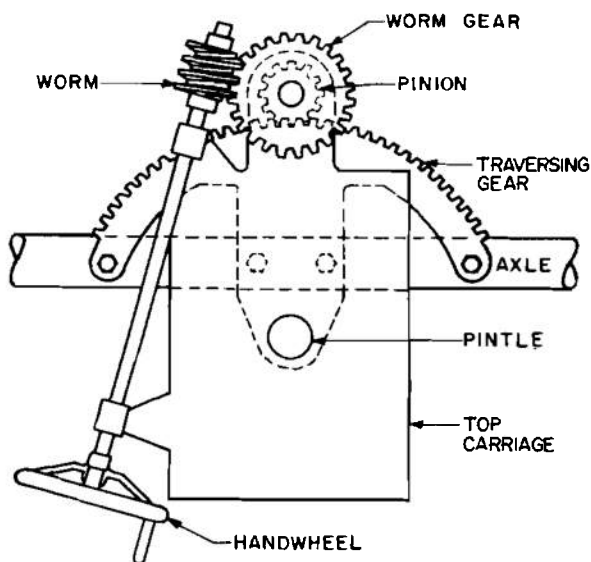Figure 2. Axle Traversing Mechanism

3

Figure 3. Pintle Traverse

Figure 4. Split Trail Mount Showing Traversing Range

2. Methods of Actuation

13. *Method A.* Power, mechanical or manual, is transmitted through a gear train whose terminal member is the traversing gear which may be either a large worm wheel or spur gear. Two installations are available. The first (Figure 5) has the driving gear on the bottom carriage and the traversing gear (driven) on the top carriage. Alternatively, the driving gear can be mounted on the top carriage and the traversing gear on the bottom carriage (refer to Figure 1). This latter arrangement permits unlimited traverse and, with the gunner moving with the traversing parts, direct sighting is more convenient. If high traversing speeds are necessary, the gear train is powered mechanically or electrically (see Figure 6). If high speeds are unnecessary or in the event of power failure, manual action is provided through a handwheel.

14. *Method B.* A variation of the pintle type traversing gear consists of a screw-nut combination which pivots on the bottom carriage near the handwheel (see Figure 7). The screw, turned by the handwheel, and nut pivot on the top carriage as it traverses about the pintle. Such an arrangement, however, is very limited in traverse range and is inherently lacking in rigidity.

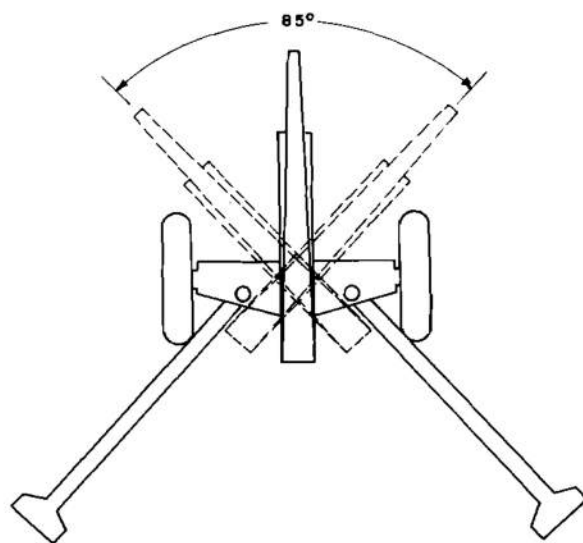15. *Method C.* Manually operated traversing mechanisms are relatively slow but rapid traverse can be achieved with those weapons having the low frictional resistance of the pintle mounting. Rapid coarse traverse is available by simply disengaging the gear train and swinging the weapon by hand until pointed in the general direction. Re-engaging the gears reverts the system to its fine traverse state.

16. *Method D.* A firing platform provides coarse unlimited traverse for a split trail weapon. This method, now outmoded, has the wheels of the weapon raised off the ground and the bottom carriage pivoted on a platform that rests on the ground. The trails are supported on a circular rail which is staked to the ground and attached to the central base with radial members. Coarse traverse is made by rotating the mount on the platform. Push bars extending from the end of the trails permit the gun crew to turn the mount manually by pushing the ends of the trails along the rail. Once in position, the trails are clamped to the rail.

C. BASE RING AND RACER TRAVERSE

17. Weapons with heavy traversing parts or which traverse while firing, require large traversing bearings to supplement a broad firing base (see Figure 8). These weapons are trained by rotating the top carriage on a base ring equipped with either ball or roller bearings. A large spur gear mounted on the
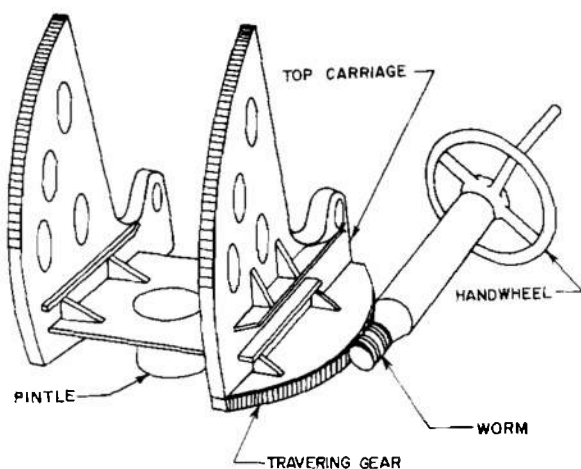
Figure 5. Traversing Mechanism With Gear on Top Carriage
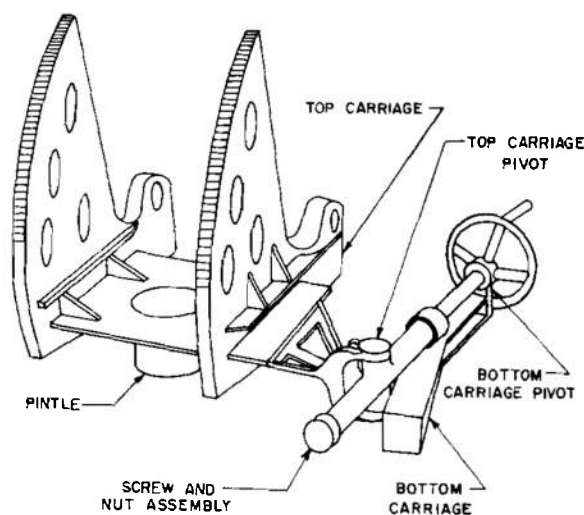
Figure 7. Screw and Nut Traversing Mechanism
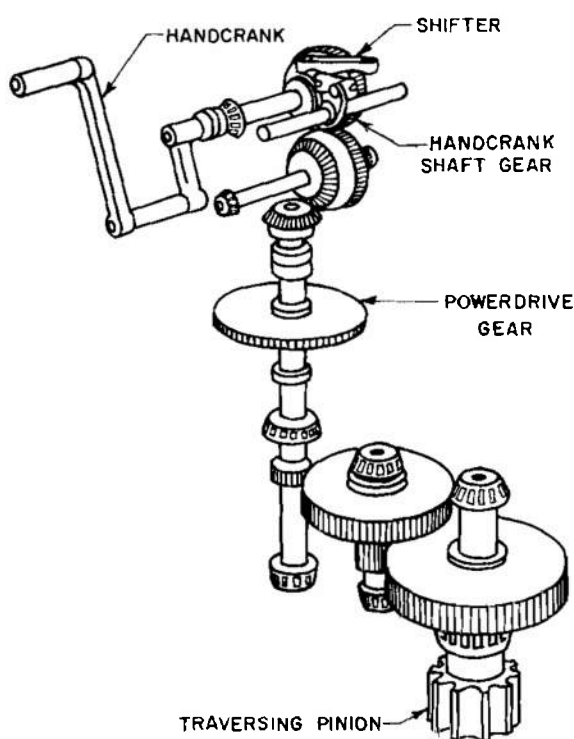
Figure 6. Powered Traversing Mechanism

Figure 8. Detail of Traversing Bearing

for rapidly traversing weapons including turret-mounted guns on tanks or motor gun carriages.

D. BALL JOINT TRAVERSE

18. By another method of traverse, the top carriage turns on a ball joint or its equivalent (see Figure 9). This method is well suited for the double recoil system as the bottom carriage provides two supports, one rear and one front (Reference 1).* The front support or turntable has the ball joint in its center. Rapid,

* References are found at the end of this handbook.

bottom carriage, concentric with the bearing ring, serves as the traversing gear. Its pinion is mounted on the traversing parts. As the pinion turns, it tracks around the traversing gear carrying with it the top carriage and cannon. This arrangement provides unlimited traverse and can be powered either mechanically or electrically. It is particularly suitable

Figure 9. Ball Joint Traversing System

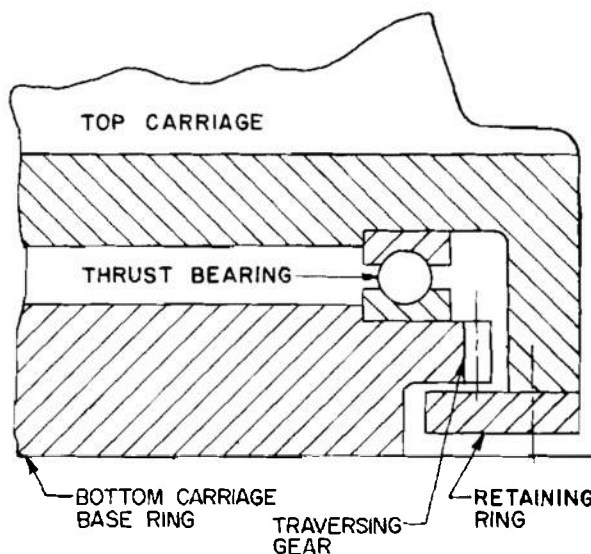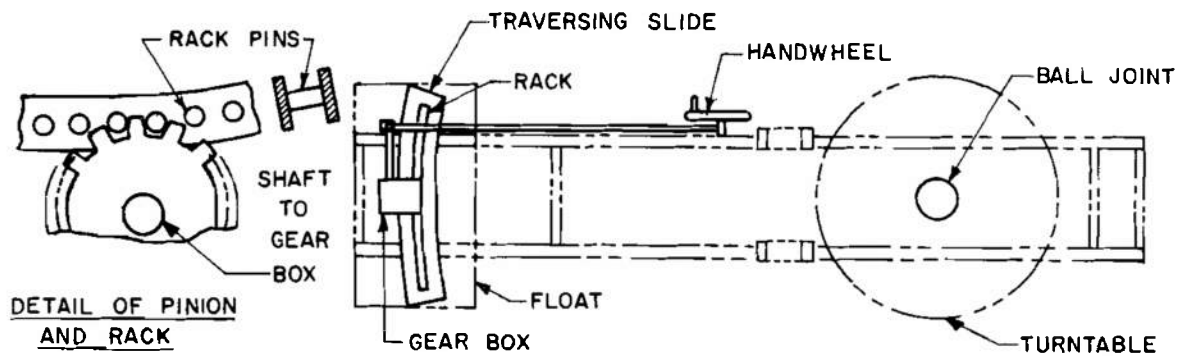unlimited coarse traverse is obtained by lifting the rear support (float) off the ground and swinging the carriage in either direction by pushing sidewise on the rear of the structure. Heavy structures are lifted with mechanical jacks which rest on circular tracks forming the outer periphery of the turntable. Light structures may be lifted by the gun crew. Fine traverse is limited, seldom exceeding ±15 degrees. The traversing mechanism, except for handwheel and shafting, is located in the rear. It is essentially the conventional gear train providing the required mechanical advantage. The rear support (float) is equipped with traversing rails on which the traversing parts slide. These rails form a circular arc the center of which is the ball joint. The traversing rack, attached to the float, also has the ball joint as its center. The rack need not have true gear teeth. Cylindrical pins may be substituted. The advantage gained here is the self-cleaning ability of pins which eliminate material that could cause jamming of full-form gear teeth.

19. The traversing mechanism for a double recoil gun is more elaborate than for a single recoil gun mounted on a similar structure. The latter has no relative motion between top and bottom carriage, therefore the gear train of the traversing mechanism is always engaged and conventional linkages are adequate. On the other hand, the double recoil gun has relative motion between top and bottom carriage. Since the traversing mechanism is attached to both, some part of the linkage or gear train must be disengaged or, where feasible, the shafting can have telescoping members. The gear train may be disengaged by separating two meshing gears, one on the top, the other on the bottom carriage (see Figure 10). The two gears separate during recoil. Just as the top carriage returns to in-battery, the two gears re-engage. Meshing is facilitated by rounding the edges of the gear teeth. To preclude excessive contact forces when the teeth do not chance to mate, one of the gears is spring loaded to permit it to slide on its shaft. Then, as long as firing continues, it does not matter if these gears are not meshed. When traverse becomes necessary, only one gear would turn but as its teeth become re-aligned with those of its mate, the spring pushes it into the properly meshed position.

20. Another method of traverse, both crude and cumbersome, positioned the entire weapon on a curved track or epi. This method was particularly adaptable to the huge railway guns, now obsolete. Unlimited traverse was available if the track formed a complete circle. This characteristic becomes naturally applicable to self-propelled weapons.

III. EQUIPMENT ASSOCIATED WITH TRAVERSING MECHANISMS

21. Several pieces of equipment are directly associated with the traversing mechanism but are units distinct enough that they cannot be considered as integral parts. For instance, power drives actuate the mechanism yet are no more a part of it than the gunner who turns the handwheel. Some, such as brakes and buffers, are considered as parts of the mount while others are classed as auxiliaries which include directors, power drives and controls,

sights, and stabilizers. Complete descriptions, design data and procedures of the latter group appear in pamphlets on fire control. The discussions here only show the relation of their functions with those of the traversing mechanism. Although the auxiliary equipment may be designed by others, those responsible for the traversing mechanism and the rest of the mount should be familiar with this equipment so that each unit may be located where it will be most effective, particularly those controls and indicators which constantly demand the attention of the gunner.

A. POWER DRIVE

22. Quick-traversing heavy guns require a sustained effort far beyond the stamina of anyone operating a manual traversing mechanism. This is especially true of high-rate-of-fire antiaircraft weapons which must keep pace with fast and elusive targets and of tanks which are constantly moving, and therefore must contend with changing target positions. Such mounts are traversed with variable-speed

power drives which are generally mounted on top carriages or in tank turrets. If units of the power drive system are mounted on both moving and stationary parts of the structure, the power and signal transmission lines must be connected by slip rings or their equivalent to preclude twisting of the lines during traverse.

23. Power drives capable of both coarse and fine traverse in either direction are electric or hydraulic, the latter being driven by an electric motor or internal combustion engine. Ultimately all power for a mobile or self-propelled weapon is derived from an internal combustion engine. (For permanent emplacements, conventional electric transmission lines may be available.) The engine drives either a generator or a hydraulic unit. If a direct drive is used only one mount can be served, but an engine-generator unit can serve several units. The engine may be attached to the mount, it may be a separate unit and borne on its own chassis, or the prime mover may be equipped with a power take-off device. When rough control can be tolerated, traverse can be achieved by gearing the engine directly to the
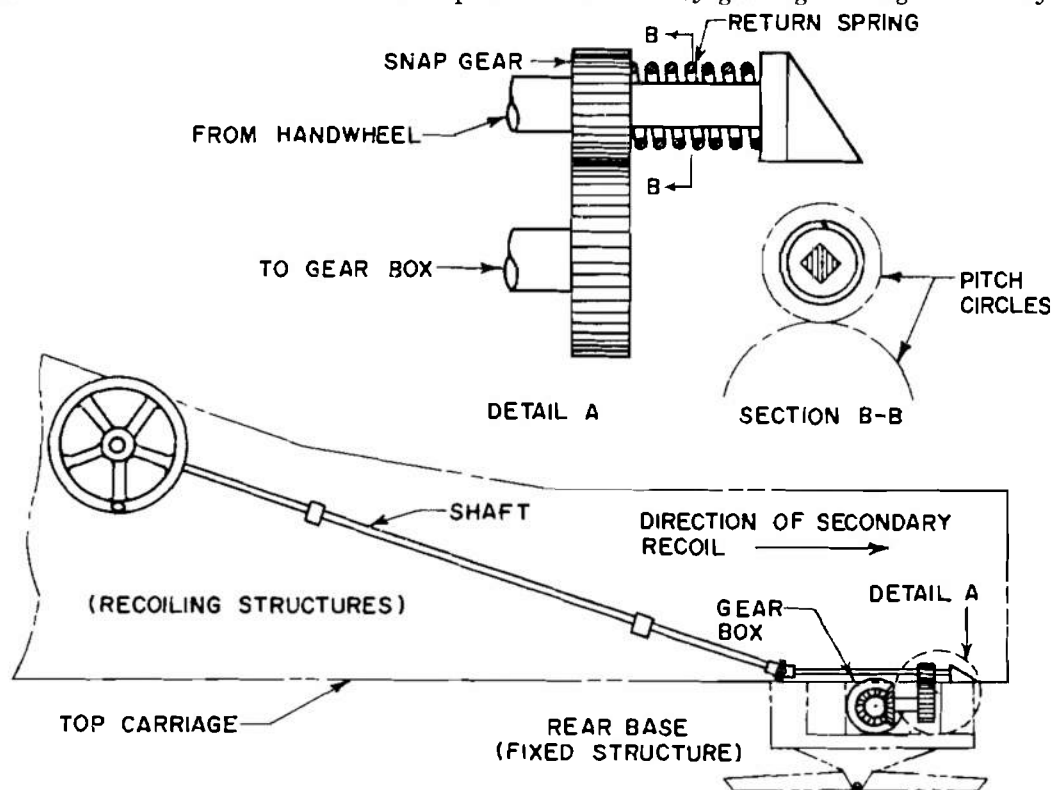

Figure 10. Arrangement of Traversing Mechanism—Double Recoil Carriage

traversing mechanism. However, this arrangement will not be suitable for fine traverse unless quick-operating and sensitive controls are provided.

24. Hydraulic drives are much more refined systems than the direct engine drive and are readily adapted to precise automatic control. The system features a hydraulic pump and motor (see Figure 11). One such system has a variable-displacement pump with a direct line to the motor. Another has a constant-displacement pump discharging into an accumulator to maintain a constant pressure. The accumulator provides flow to the motor. Motor speeds and direction, hence traversing speeds and direction, are regulated by controlling the flow of the variable-displacement pump or by a valve controlling the flow through two lines leading to the motor.

25. Electric power drives behave similarly to hydraulic drives. They can traverse in either direction at variable speeds and are readily adaptable to precise automatic control. These characteristics are made available through an amplidyne system whose voltage output to the drive motor can vary according to the traversing speed requirements.

B. DIRECTORS AND OTHER CONTROLS

26. Combining the information of its tracking devices with that received from range finders, directors compute firing data electronically and transmit these data as signals to the guns. Sending and receiving apparatus are synchronized with the traversing system. In a totally automatic installation, the difference in signals (called the error signal) between those generated in the director and those on the gun corresponds to the off-target position of the gun. Responding to the signal, the power drives traverse the gun until the error disappears, indicating on-target position. In installations not totally automatic, the gunner controls the traversing operation. He is guided by a pointer on a dial which indicates the firing position as determined by the director. A second pointer on the dial, synchronized with the traversing mechanism, represents the actual gun position. To bring the gun into firing position, the gunner merely traverses
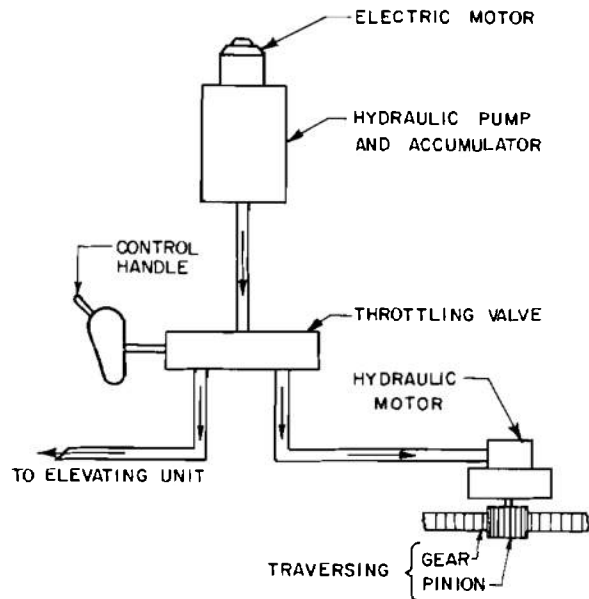
Figure 11. Hydraulic Drive

the weapon until the second pointer matches the first thus indicating a position which corresponds with the firing data.

27. There are several methods of controlling power traverse without the aid of directors. These are basically the same, differing only in the method of manipulation. One method has control by handwheel with the drive operating as a power assist. Another has handles turning on a vertical axis similar to a steering wheel. Still another has a "joy stick" arrangement. Finally, control may be achieved by manipulating a ball which serves as the initiating component of the control unit.

28. With the help of Figure 12, the mechanics of a control unit are discussed in some detail to illustrate a typical unit. Aside from the ball and gear trains, the controller has these seven basic components:

A. Magnetic clutch, direct engagement between ball and G.
B. Centering cam, renders G inoperative when clutch is disengaged.
C. Cam return arm, returns cam to neutral position.
D. Differential, provides access of gear train to E.
E. Synchro transmitter, sends operating signals to power drive.

F. Constant speed motor, provides continuous motion to G.

G. Ball-disk integrator, regulates traversing speed.

Direct manual control is exercised by disengaging the clutch A, thus directing all motion from the ball through gears 2 and 3 of the differential D to the synchro transmitter E. Gear 1 is held stationary. Whenever the control ball rotates the friction roller, the traversing parts are turned in the direction of and at a rate proportional to that of the ball. In the meantime, the spring force on the cam return arm C rotates the centering cam B into its centering position which in turn moves the integrator G into idle. Here balls 6 are centered on the disk 5 of the integrator where no peripheral motion exists. No motion can now be transmitted to the gear train linking integrator to differential.

29. Assisted manual control becomes available when the clutch A is engaged. The gear train between control ball and integrator G is uninterrupted. The control ball now exercises control of the traversing direction and speed merely by its own displacement which regulates the displacement of yoke 8 of the integrator. As the yoke moves in either direction from the axis of the constant speed disk 5, it carries balls 6 toward faster peripheral motion. The balls transmit this motion to roller 7 and through the gear train to 1 of the differential and eventually to the synchro transmitter E. Gear 2 of the differential D is held stationary by the motionless control ball. Traverse is reversed by moving the balls of the integrator across the disk axis and is stopped by reversing the control ball until the cam centers in its neutral position or by releasing the clutch.

C. STABILIZERS

30. The pitching and slewing of a moving tank disturbs gun accuracy by creating an unstable firing base which would require the vehicle to come to a halt before firing begins if compensating measures were not provided. During combat, such maneuver would waste time and increase vulnerability to an intolerable extent. Compensations for erratic motion are provided by stabilizers whose primary function is to maintain the hold of a gun on its target. Gyroscopes are well suited for this purpose. To complete the stabilizer system, some device, often hydraulic or electric, must be provided to convert gyroscope reactions into signals and to transmit them to the traversing system which then responds accordingly and corrects azimuth misalignment between gun and target.

IV. DESIGN REQUIREMENTS

A. GENERAL DESIGN DATA

31. Four basic design requirements are essential to any traversing mechanism, namely; control, power transmission, precision, and sensitivity. Control is exercised by two mediums; the fire control equipment which aims

A - MAGNETIC CLUTCH
B - CENTERING CAM
C - CAM RETURN ARM
D - DIFFERENTIAL
E - SYNCHRO TRANSMITTER
F - CONSTANT SPEED MOTOR
G - BALL - DISK INTEGRATOR

Figure 12. Ball Drive Controller

the weapon but is not considered part of the mechanism, and the limit stops and locking device which maintains the aim with respect to azimuth position. The locking device, considered a part of the traversing mechanism, may be a brake, a clutch, or an irreversible screw. Limit stops, usually hydraulic buffers, prevent overtravel. Power, whether manual or mechanical, generally is transmitted by a gear train. Precision depends on the quality of manufacture, particularly in relation to backlash, while sensitivity involves the gear ratio. These requirements plus the location of all components become the basis of traverse mechanism design.

32. The gear train, no matter how simple, is the principal part of the traversing mechanism and must satisfy all basic design requirements. Its gear ratio should be considered first. Not only does it prescribe the effort at the source, it also limits sensitivity. A high gear ratio means slow traverse with respect to applied motion; a low ratio means correspondingly faster traverse. Although a high gear ratio demands a low torque output from the power source, the power requirements alone are unaffected. For example, assume that the torque at the traversing gear, incorporating the efficiency of the gear train, is 2400 lb-ft. For slow traverse, a gear ratio of 600:1 reduces it to 48 lb-in at the handwheel; a ratio of 300:1 reduces it to 96 lb-in. For fast traverse, say 30 degrees per second, the shaft output requires $2\frac{3}{4}$ horsepower whether the motor turns at 3600 rpm through a gear reduction of 600:1 or at 1800 rpm through a reduction of 300:1. Specified handwheel effort and motor characteristics will establish the gear ratio. Thereafter, type, number, size, and location of gear train components may be determined.

33. The design of the gear train involves some trial and error procedure, more so if mechanically powered because preliminary estimates of both efficiency and inertia must be made whereas only efficiency need be estimated for the manually operated train. Final computed data must agree reasonably well with the estimates before the preliminary results are acceptable. In short, the effort required to move the traversing parts, including

the moving components of the traversing mechanism, must not exceed the available effort at the power source.

34. Design data are computed from statics and dynamics of the moving masses. Data on statics include the weight moment about the traversing axis and the torque due to frictional resistance of the traversing bearing. If the weapon is on a slope and the center of gravity of the traversing parts is not on the axis (see Figure 13), the weight moment is

$$M_w = W_{tr}R_t \sin \theta_t \cos \phi_a \qquad (1)$$
where

W_{tr} = weight of traversing parts
R_t = radius, traversing axis to CG
θ_t = slope of terrain
ϕ_a = location of CG with respect to horizontal line in plane parallel to the slope

35. On level terrain or if leveling devices are used, θ_t and therefore M_w are zero. Or, should the weapon, particularly a tank, be equipped with an azimuth equilibrator, the weight moment is balanced and effectively reduced to zero. Otherwise M_w may become appreciable, reaching a maximum when ϕ_a equals zero. The
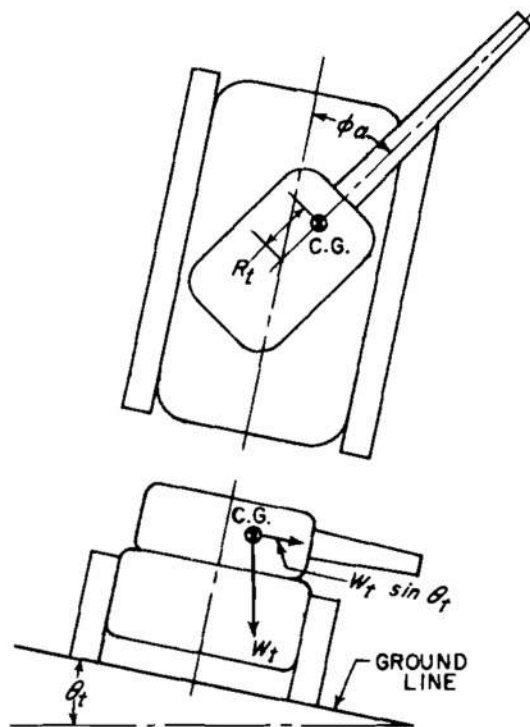

Figure 13. Weight Moment Diagram

10

other static component of the torque forms in the traverse bearing and is simply

$$T_b = \mu F_{br} R_\mu \qquad (2)$$

where

μ = coefficient of friction

F_{br} = total normal force on the traversing bearing

$R_\mu = \dfrac{2}{3}\dfrac{r_o^3 - r_i^{3*}}{r_o^2 - r_i^2}$, frictional radius (2a)

r_i = inside radius of bearing

r_o = outside radius of bearing

Due to the presence of recoil forces, F_{br}‡ increases appreciably thereby increasing the torque, T_b. The additional torque must be considered if the weapon traverses during the recoil cycle.

36. Another component of the torque at the traversing gear, the firing couple, is generated during firing and is due to the eccentricity of the recoiling parts about the traversing axis. (Two guns, mounted side by side on the same carriage, will produce a similar torque if not fired simultaneously.) According to Figure 14, the component of the firing couple affecting traverse is

$$T_f = (aF_g - bF_a)\cos\theta \qquad (3)$$

where

F_a = inertia force of recoiling parts

F_g = propellant gas force

θ = angle of elevation

Careful design should hold this torque to a minimum, the object being to have bore center, traversing axis, and mass center of the recoiling parts lie in a vertical plane. This alignment is not always possible. Space limitations and required structural locations may cause an unequal distribution of weight and create an unbalance about the bore axis. And, regardless of the care exercised, manufacturing tolerances may augment this unbalance. The inertia forces of the unbalanced weight produce the firing couple. If the traversing gear train is subjected to the firing torque, the effect on

___

* Reprinted by permission from *Analytical Mechanics for Engineers* by F. B. Seely and N. E. Ensign, copyright 1952, John Wiley & Sons, Inc., page 143.

‡ The method for computing the bearing load is presented in Reference 2, Chapter VIII.
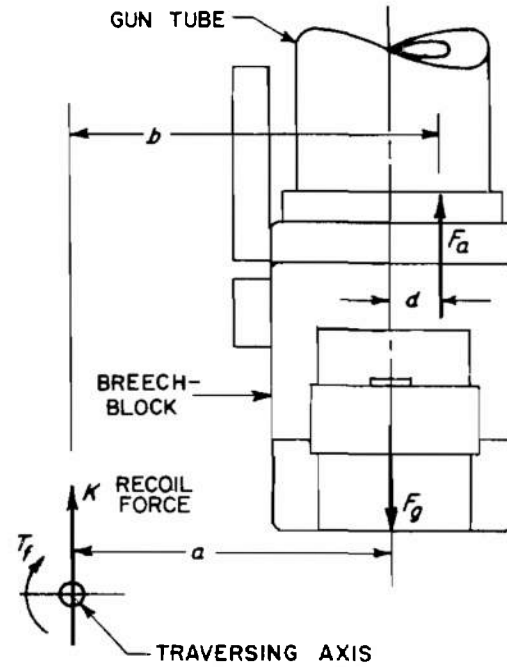
Figure 14. Firing Couple Diagram

the power supply may be significant. If locked out, only the gears between traversing gear and locking device will be disturbed.

37. This couple can be modified to a degree by shifting the traverse axis. No location can sustain a zero torque throughout the firing cycle. This is especially true during the propellant gas period as the gas force varies from a maximum to zero. For example, consider a hypothetical gun whose recoil force is unity and the propellant gas force ranges from 16 to 0, thus

$$K = 1$$
$$F_a = F_g - K = F_g - 1 \qquad (3a)$$

Using the geometry of Figure 14 and assuming horizontal firing when $a = 0$, the firing couple varies from $T_f = -15d$ to $T_f = d$. When zero torque is desired at maximum gas pressure,

$$a = d\,\frac{F_a}{F_g - F_a} \qquad (3b)$$

$$a = d\,\frac{15}{16 - 15} = 15d \qquad (3c)$$

$$b = a + d = 16d \qquad (3d)$$

The firing torque, as F_g reduces to zero, varies from $T_f = 0$ to $T_f = 16d$ as shown in Table 1. Now assume an intermediate value, $a = 7d$

11

and $b = 8d$. Firing torque variations for these values are listed in Table 2.

TABLE 1. *Firing Torque With Maximum Displacement of Traverse Axis*

F_g	F_a	aF_g	bF_a	T_f
16	15	$240d$	$240d$	0
15	14	$225d$	$224d$	$1d$
14	13	$210d$	$208d$	$2d$
..	..	..	..	..
..	..	..	..	..
2	1	$30d$	$16d$	$14d$
1	0	$15d$	0	$15d$
0	-1	0	$-16d$	$16d$

TABLE 2. *Firing Torque With Intermediate Displacement of Traverse Axis*

F_g	F_a	aF_g	bF_a	T_f
16	15	$112d$	$120d$	$-8d$
15	14	$105d$	$112d$	$-7d$
14	13	$98d$	$104d$	$-6d$
..	..	..	..	..
..	..	..	..	..
9	8	$63d$	$64d$	$-1d$
8	7	$56d$	$56d$	0
7	6	$49d$	$48d$	$1d$
..	..	..	..	..
..	..	..	..	..
2	1	$14d$	$8d$	$6d$
1	0	$7d$	0	$7d$
0	-1	0	$-8d$	$8d$

38. These examples show a varying firing torque during the propellant gas period no matter where the traversing axis is located. After this period only the inertia forces induced by the recoil mechanism are acting and for a considerably longer time than the gas period. The obvious solution here is to locate the traversing axis in the plane of the recoiling mass center. Except for the gas period, the firing torque is now always zero. But, as mentioned previously, this position may not be readily available or easily located while in the design state. Under such conditions, it is advisable to locate the traverse axis in the plane of the bore center inasmuch as the maximum torque appears only briefly and the sustained load is considerably smaller than other conditions, two of which are illustrated in Tables 1 and 2.

39. The fourth component of the torque at the traversing gear is that required to accelerate the traversing parts.

$$T_a = \Phi\alpha \tag{4}$$

where

Φ = mass moment of inertia of the traversing parts about the traversing axis

α = maximum traversing acceleration

The general expression for maximum torque at the traversing gear is

$$T_R = M_w + T_b + T_f + T_a \tag{5}$$

Recapitulate, with emphasis on the various conditions. For a level weapon traversed manually

$$T_R = T_b \tag{6}$$

For the same weapon on a slope

$$T_R = M_w + T_b \tag{7}$$

For a level weapon traversed by mechanical or electrical power

$$T_R = T_b + T_a \tag{8}$$

For the same weapon on a slope

$$T_R = M_w + T_b + T_a \tag{9}$$

B. GEAR TRAIN

40. The gear train, as a mechanical transformer, changes the large traversing gear torque to a lesser value at the power source. For a handwheel operated mechanism the ratio of the two values is the gear ratio modified by the efficiency (see Figure 15). The gears have even numbers, the pinions, odd. Beginning with the traversing gear 6, trace the torque through the system by converting it to gear tooth load and back again to torque, taking the efficiency, η_g, into account at each mesh. Thus the gear tooth load between gears 5 and 6 is

$$F_{56} = \frac{T_R}{R_{p_6}} \tag{10}$$

The torque in gears 4 and 5 becomes

$$T_{45} = \frac{1}{\eta_g} R_{p_5} F_{56} = \frac{1}{\eta_g} T_R \frac{R_{p_5}}{R_{p_6}} \tag{11}$$

The gear load between gears 3 and 4 is

$$F_{34} = \frac{T_{45}}{R_{p_4}} = \frac{1}{\eta_g} T_R \frac{R_{p_5}}{R_{p_4} R_{p_6}} \tag{12}$$

12

The torque in gears 2 and 3 becomes

$$T_{23} = \frac{1}{\eta_g} R_{p_3} F_{34} = \frac{1}{\eta_g^2} T_R \frac{R_{p_3} R_p}{R_{p_4} R_{p_6}} \quad (13)$$

The gear load between gears 1 and 2 is

$$F_{12} = \frac{T_{23}}{R_{p_2}} = \frac{1}{\eta_g^2} T_R \frac{R_{p_3} R_{p_5}}{R_{p_2} R_{p_4} R_{p_6}} \quad (14)$$

And, the torque at gear 1 or at the power source becomes

$$T_t = \frac{1}{\eta_g} R_{p_1} F_{12} = \frac{1}{\eta_g^3} T_R \frac{R_{p_1} R_{p_3} R_{p_5}}{R_{p_2} R_{p_4} R_{p_6}} \quad (15)$$

This is the torque required at the power source to turn the traversing parts. Expressed generally

$$T_t = \frac{1}{\eta_g^n} \frac{T_R}{r_g} \quad (16)$$

where

n = number of gear meshes

r_g = gear train ratio

η_g = 0.98 to 0.99*, efficiency of each spur or bevel gear mesh

When a pinion is replaced by a worm whose efficiency is η_w

$$T_t = \frac{1}{\eta_w \eta_g^{n-1}} \frac{T_R}{r_g} \quad (17)$$

For the worm being the driving member

$$\eta_w = \frac{\cos \beta - \mu \tan \lambda\ddagger}{\cos \beta + \mu \cot \lambda} \quad (18)$$

where

β = gear pressure angle

λ = lead angle of worm gear

μ = coefficient of friction

When frictional losses in the thrust bearing of the worm are considered, the efficiency, based on Equation (15-11) of Reference 4, is

where

D_b = effective diameter of the thrust bearing

D_w = pitch diameter of worm

μ_b = coefficient of friction of the thrust bearing

Expressed in terms of the composite efficiency of the gear train

$$T_t = \frac{1}{\eta} \frac{T_R}{r_g} \quad (19)$$

41. The inertia of the gear train contributes to the total required effort for motorized traversing mechanisms. As the train is being accelerated, the torque on each gear progresses through the train similarly to the one coming from the traversing gear. Referring to Figure 15, start at the integral gear 45 and express all accelerations in terms of the traversing acceleration.

$$T_\alpha = \frac{1}{\eta_g^{n-1}} T'_{45} \frac{R_{p_1} R_{p_3}}{R_{p_2} R_{p_4}}$$
$$+ \frac{1}{\eta_g^{n-2}} T'_{23} \frac{R_{p_1}}{R_{p_2}} + \frac{1}{\eta_g^{n-3}} T'_1 \quad (20)$$

but

$$T'_{45} = \Phi_{45} \alpha_{45}, \quad T'_{23} = \Phi_{23} \alpha_{23}, \quad T'_1 = \Phi_1 \alpha_1$$

and

$$\alpha_{45} = \alpha \frac{R_{p_6}}{R_{p_5}}, \quad \alpha_{23} = \alpha \frac{R_{p_4} R_{p_6}}{R_{p_3} R_{p_5}}, \quad \alpha_1 = \alpha \frac{R_{p_2} R_{p_4} R_{p_6}}{R_{p_1} R_{p_3} R_{p_5}}$$

therefore, through substitution

$$T_\alpha = \left(\frac{1}{\eta_g^3} \Phi_{45} \frac{R_{p_1} R_{p_3} R_{p_6}}{R_{p_2} R_{p_4} R_{p_5}} \right.$$
$$+ \frac{1}{\eta_g^2} \Phi_{23} \frac{R_{p_1} R_{p_4} R_{p_6}}{R_{p_2} R_{p_3} R_{p_5}} + \frac{1}{\eta_g} \Phi_1 \frac{R_{p_2} R_{p_4} R_{p_6}}{R_{p_1} R_{p_3} R_{p_5}} \right) \alpha \quad (21)$$

For a conservative estimate of this torque, assume the gear train efficiency for all components rather than the summation of the

$$\eta_w = \frac{\cos \beta - \mu \tan \lambda}{\cos \beta \left(1 + \mu_b \dfrac{D_b}{D_w} \cot \lambda \right) + \mu \cot \lambda \left(1 - \mu_b \dfrac{D_b}{D_w} \tan \lambda \right)} \quad (18a)$$

---

* Reference 3, pages 320, 346.

‡ Reference 4, page 389, Equation (15-8).

nized as the design approach to the servo-system. The power source must be capable of developing peak torque at peak acceleration while the mount is traversing at peak speed. Investigations have shown that optimum power requirements and speed reductions can be determined to fit the particular need of a traversing system. Some of these relationships are expressed in Equations 24, 25 and 26.*

$$r_g = \gamma r_n \tag{24}$$

$$\gamma = \frac{1}{2}\frac{t_l}{t_m} \tag{25}$$

$$\gamma P_m = 2P_l \tag{26}$$

where

P_m = power generated by motor

P_l = power required to rotate traversing parts

$r_n = \dfrac{\text{peak speed rating of motor}}{\text{peak rotational speed of mount}}$

r_g = required gear train ratio

t_m = time constant of motor

t_l = time constant of traversing parts

$\gamma = \dfrac{\text{maximum operating motor speed}}{\text{peak motor speed}}$

The time constant of the motor is defined as the time required to bring the motor, running free of load, from standstill to maximum operating speed at maximum angular acceleration. The time constant of the traversing parts is defined as the time required to bring these parts to maximum speed from a standstill at maximum acceleration. The above expressions will serve as a guide for the traversing mechanism designer who, although not necessarily the power drive designer, must compile the essential design data, meanwhile keeping in touch with the power drive specialist to make sure that these data are not too demanding.

43. The maximum traversing acceleration depends on the tactical use of the weapon. For instance, field artillery fires on fixed targets, thereby permitting minutes rather than fractions of a second for aiming; whereas guns firing on moving targets must have high acceleration to acquire and track the target quickly. Thus, tank turrets and antiaircraft guns must have appreciable angular acceleration, on the order of 0.5 rad/sec². Tanks, while maneuvering, slew or pivot at angular acceleration far in excess of the traversing acceleration of 0.5 rad/sec². To preclude relative motion between turret and hull, the traversing mechanism must serve as a travel lock to form a fixed link between them. Thus the gear train will transmit the torque from hull to turret. This torque is limited to that induced by a maximum acceleration of 6 rad/sec². Should hull accelerations exceed this value, a clutch, somewhere in the gear train, slips at the corresponding torque and permits relative motion between hull and turret.

44. Assuming that the gear ratio has been established, type, number, size, and location of the gears are determined next. The relative positions of the traversing gear and handwheel or power drive, in addition to space availability, locate the gears. Accessibility for maintenance also influences the location. The gear train should be protected by a gear box or other suitable cover. Provision for ready access should be made for cleaning and lubricating. For parallel shafting, spur, helical, or herringbone gears are available. The helical and herringbone gears are smooth running at high speeds and are comparatively low stressed, with the herringbone having the added advantage of neutralizing the induced end thrust of the helical gear. Bevel and worm gears are used for nonparallel shafting, the latter being particularly adapted for high speed or high torque ratios. Gears should be few in number and small to minimize inertial and frictional losses. They must mesh with a minimum of clearance, the total backlash in the train being limited to about one mil. Backlash is sometimes eliminated with an antibacklash device. Tooth loads are important in design but other factors such as speed and accuracy of tooth form must also be considered. However, detailed design procedures are available in most good machine design texts.‡ Also, specialists are always within reach at gear plants for consultation.

---

* Reference 5.

‡ Such as Reference 4.

C. HANDWHEELS

45. Manual operation of a traversing mechanism begins with a crank or handwheel. Two-handed effort is conducive to smoother and more uniform operation and can be arranged provided that one hand is not needed to manipulate other controls simultaneously. The smoother action will be less tiring. For two-handed operation, a double crank (see Figure 6) or a double handwheel (Figure 16) serves the purpose. Handwheels, other sighting controls, and associated indicators must be located near each other for ready access and easy observation. All manipulating equipment such as wheels, handles, levers, and push buttons must be large enough to be gripped easily with gloved hands during cold weather.

46. Handwheels are designed on the basis of the physical effort required to overcome static loads and frictional resistance in the traversing bearing and other components of the traversing system. Dynamics need not be considered as speeds are necessarily low. Handwheels are normally circular with a handle and a counterweight attached opposite to each other on the rim. Any structural material may be used including cast iron, steel, aluminum, and magnesium. Some are cut from plate stock, others are cast. Some work has been done to determine a man's capacity for turning a handwheel. His best effort for sustained operation was achieved while producing 50 lb-in of torque at a turning radius of 7 inches.* For equipment that cannot meet these limits, handwheel force should not exceed 12 pounds. Where traversing loads are extreme, the mechanical advantage of the gear train must be increased to reach the required handwheel effort. These same conditions prevail for handwheels on mechanically powered systems.

D. ANTIBACKLASH DEVICE

47. To be effective, a weapon must be accurate and its accuracy depends to a great extent on the precision of its traversing mechanism. Loosely meshed gears cannot be

* Reference 1.
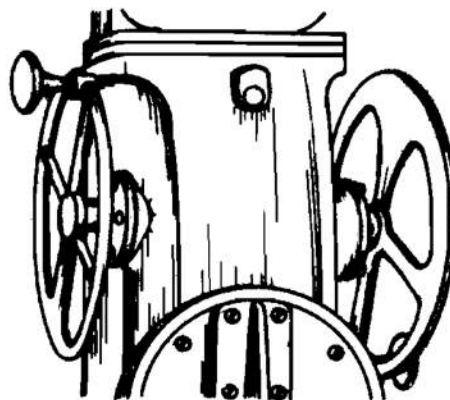
Figure 16. Double Handwheel Mechanism

tolerated although a small amount of backlash is necessary to reduce friction, to compensate for thermal expansion, and to allow for manufacturing tolerances. Careful machining will hold backlash to a minimum. Or, gears can be assembled tightly or even with a slight interference and then "run-in." For small gears, light oil is generally used for the run-in process. For large gears, a mixture of household cleanser and water or one of grinding compound and oil may be used. If the cost is justified, some gears may be mounted on adjustable centers. After being fitted, the assembly is doweled into place. However, even these techniques do not eliminate backlash indefinitely; for, wear will eventually cause some play between the gear teeth.

48. Antibacklash devices eliminate play between gear teeth. These devices work on the principle of preloading the last gear in the train in two directions so that tooth contact is always present regardless of the direction of motion. Figure 17 has two spring-loaded, concentric gears, A and B, on the same shaft. A is keyed; B is free to turn but both mesh at the same point with gear C. The springs induce a torque tending to turn the gears in opposite directions, thus loading two adjacent teeth on C in opposite directions. Backlash is precluded only when the induced spring torque is larger than the applied torque of gear A. Because of limited spring loads, this type is suited only for small gearing and is used extensively in indicator gear trains.

49. Antibacklash devices requiring relatively light spring loads are also available for large

gearing (see Figure 18). For this type, coil spring provides an axial force to load the helical gear serving each pinion. The gear teeth convert the small axial load into large tangential components which provide the torque necessary to keep the traversing pinions in contact with the traversing gear at all times. The torque at the spring-loaded pinion is

$$T_s = \frac{F_s}{\tan \lambda} R_p \qquad (27)$$

where

F_s = spring load
R_p = pitch radius of the spring-loaded pinion
λ = helix angle

To be effective, this torque must exceed that applied through the traversing mechanism by the power source.

E. LOCKING DEVICES

1. Worm Gear

50. A locking device performs two functions. First, as a control measure, it holds the traversing parts in a fixed position during firing by resisting any unbalanced couples tending to rotate the traversing parts. Second, as a safety measure, it prevents these same couples from reversing through the gear train to spin the handwheel thus endangering personnel. The simplest locking device may be a component of the gear train, namely, the irreversible
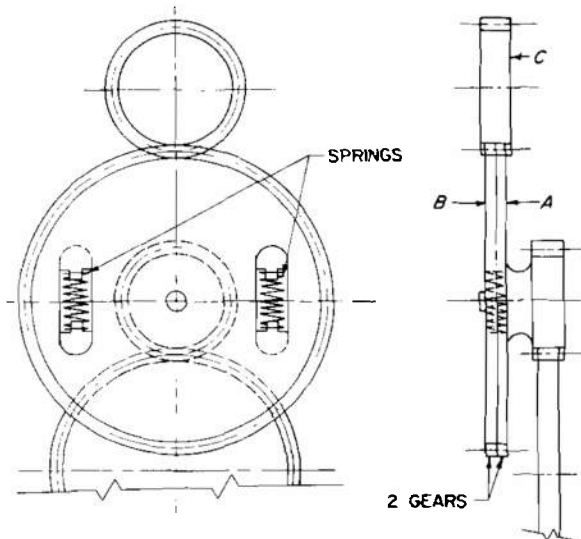
Figure 17. Antibacklash Device, Tangential Spring Type
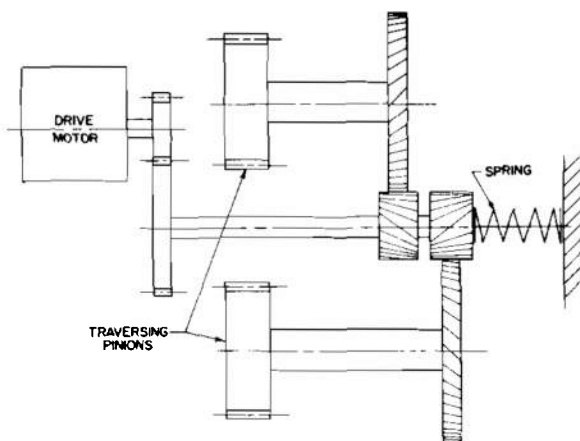
Figure 18. Antibacklash Device, Axial Spring Type

worm. It should be near the traversing gear in the gear train sequence to protect gears which are designed for lighter loads from the excessive loads that occur in tanks.

2. Brakes

51. A gear train without an irreversible feature such as the self-locking worm must rely on brakes to hold the traversing parts. Mechanical, electric, or electro-hydraulic ones may be used. The last two, when equipped with interlocks, add safety to gun handling. For instance, limit switches are so located that the brakes are applied just as the traversing gun reaches obstructions in the firing path. Brakes also hold the gun in position while it is being loaded. The brake may appear anywhere in the gear train, its capacity being governed by its proximity to the traversing gear. A large brake is needed when it is adjacent to this gear where torques are high. A relatively small one is needed next to the power supply where torques are low. If near the traversing gear, it will relieve the gear train of some dynamic loads. Cost comparisons and availability of space determine whether brakes are to be large and gears small or whether brakes are to be small and the gears relatively large.

3. Smith No-Bak Device

52. Power driven mounts require reversibility but this reversibility should not be ex-

tended to the handwheel whose inadvertent spinning may prove dangerous. A special coupling called a Smith No-Bak Device transmits torque in either direction from handwheel to gear train but precludes a reversal of this activity. Figure 19 illustrates the device. The shaft is divided into two parts. Driving and driven shaft have overlapping ends, a segment extending past the diameter being cut from each shaft. The void thus formed makes room for the spring-loaded locking bar, the shear member transmitting the handwheel effort. Shafts and bar rotate inside the lock ring which is keyed to the housing. The locking bar with ends slightly off center, bears against the inner surface of the ring on the drive shaft side of the diameter and is held firmly between the shafts by a flat spring. It rotates with them in the direction of the applied torque. Torque from the handwheel moves the locking bar toward the center, releasing its contact with the lock ring and permitting both shafts to turn. Torque from the other end holds the bar off center causing it to jam against the lock ring, stopping all motion. A slip clutch in the gear train between the No-Bak and the traversing parts yields to the torque to eliminate the No-Bak as a lock for the whole traversing system.

4. Clutches

53. Clutches are used either to link a power source to a gear train or to control the amount of torque transmitted. Positive types such as square jaw clutches are recommended for systems which are stopped during clutch operation. This type is small for the power it is capable of transmitting and construction costs are low. Friction types including cone, disk, and ring clutches, are recommended for systems in motion so that torque may be applied gradually thus reducing the possibility of sudden loading. Slip clutches, a type of friction clutch, transmitting only a limited torque, are used in Smith No-Bak installations and in traversing systems of tanks to protect the traversing mechanism from large reverse torques. Much more sensitive than brakes or other clutches, the slip clutch relies upon a fairly constant frictional resistance. Since the coefficient of friction of its rubbing surfaces is influenced by atmospheric conditions, its performance may become erratic, therefore the undesirable characteristic must be included as a design consideration. If the slip clutch is installed near the source of the reverse torque, the gear train does not become burdened with its own inertia. The advantage of lower gear
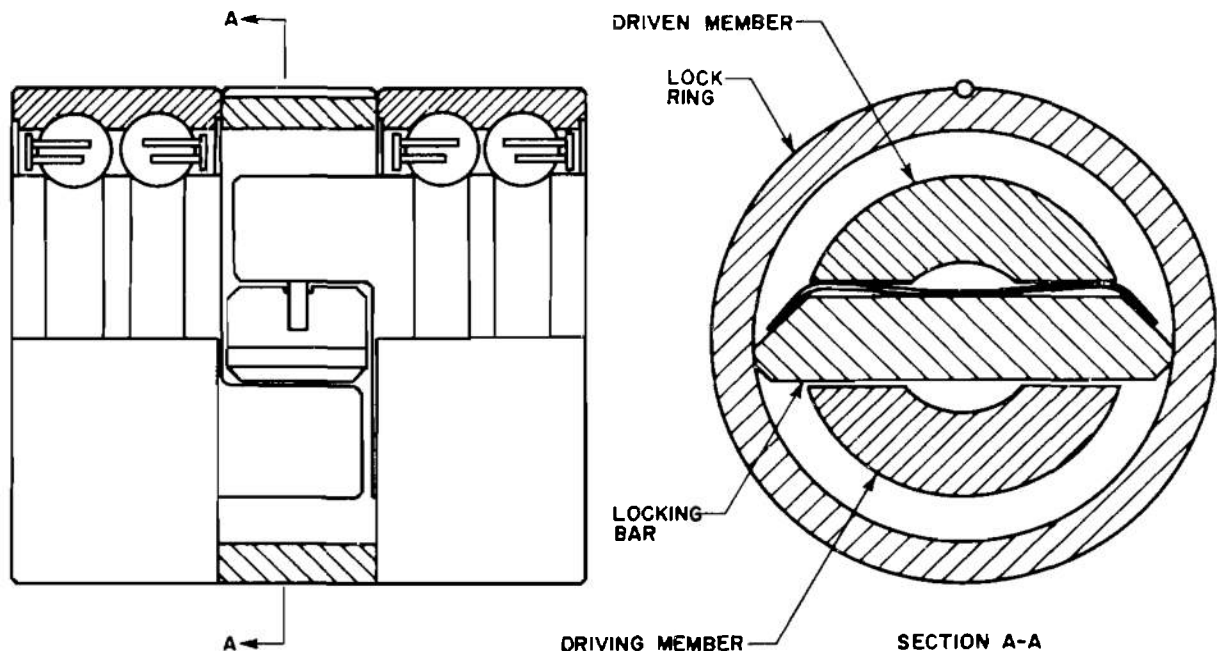

Figure 19. Smith No-Bak Device

loads obtained here is offset by the large and costly clutch required to transmit a large operating torque. On the other hand, if placed near the low-torque, high speed end of the gear train, a much smaller and less costly clutch is needed to generate the required low frictional resistance. The advantage gained here more than compensates for the somewhat larger capacity gears required to carry the additional inertial loads.

54. The recommended approach for the design of brakes and clutches parallels that for gears. Good texts which contain detailed design procedures are readily available.* One should be thoroughly familiar with the procedures in order to select suitable components for the initial design concepts. Art, as well as science, is involved, particularly for slip clutches; therefore, if the traverse mechanism designer is not expert in this field, a specialist should be consulted before designs are finalized.

5. Buffers

55. The traversing range of many weapons is limited primarily because of tactical use. Single trail or split trail field weapons firing on fixed targets need only a small traversing range, primarily for fire correction. In weapons not designed for all-round fire provision the traverse must be limited to restrict the line of action of the recoil forces to an area within the stable region of the weapon. On power driven units, limit switches give the signal for torque reversal. Handwheel operation can be stopped at will. But neither provides a positive stop. On small, manually traversed mounts where inertias are low, a filled-in tooth at each end of the traversing gear or mating lugs on top and bottom carriage serve as solid stops. On large or power driven mounts where inertias are high, some shock absorbing device must be used to cushion the impact and absorb the energy. Springs are unsatisfactory because of their ability to store energy and bounce back. Hydraulic buffers are ideal because of their energy absorbing characteristics. These buffers are not completely hydraulic because a light

spring is needed to return and hold the moving parts in their stand-by position.

56. Buffers bring the moving parts to a firm but smooth stop and should be designed for the severest condition. Although limit switches stop the power supply and may even reverse the torque at the power source, the buffers should, in the event of signal failure, be capable of absorbing the kinetic energy of the traversing parts with the driving torque fully applied. Maximum efficiency is achieved at a constant buffer force of

$$F_b = \frac{E_t}{S_b} + \frac{T_R}{R_b} \qquad (28)$$

where

E_t = maximum kinetic energy of traversing parts
R_b = radius, traversing axis to buffer line of action
S_b = total buffer stroke
T_R = applied torque of traversing gear

57. Restriction of oil flow through a variable orifice develops the resisting force. One method for varying the orifice is shown in Figure 20. Here two grooves, located 180 degrees apart for balance, provide the necessary restriction. Orifice areas, largest at the beginning of the stroke where velocities are highest, gradually grow smaller along the stroke as velocities decrease and eventually become zero. This area at any position along the stroke is

$$a_o = \frac{v_x}{c_o}\left(\frac{\rho A_b^3}{2F_b}\right)^{1/2} \ddagger \qquad (29)$$

where

A_b = effective area of buffer piston
c_o = orifice coefficient
F_b = buffer force
v_x = buffer velocity at any position, x, of stroke
ρ = mass density of hydraulic fluid

Once the orifice area is established, any change in the dynamics of the traversing parts will automatically change the buffer force and velocity which, according to Equation 29,

---

* Such as Reference 4.

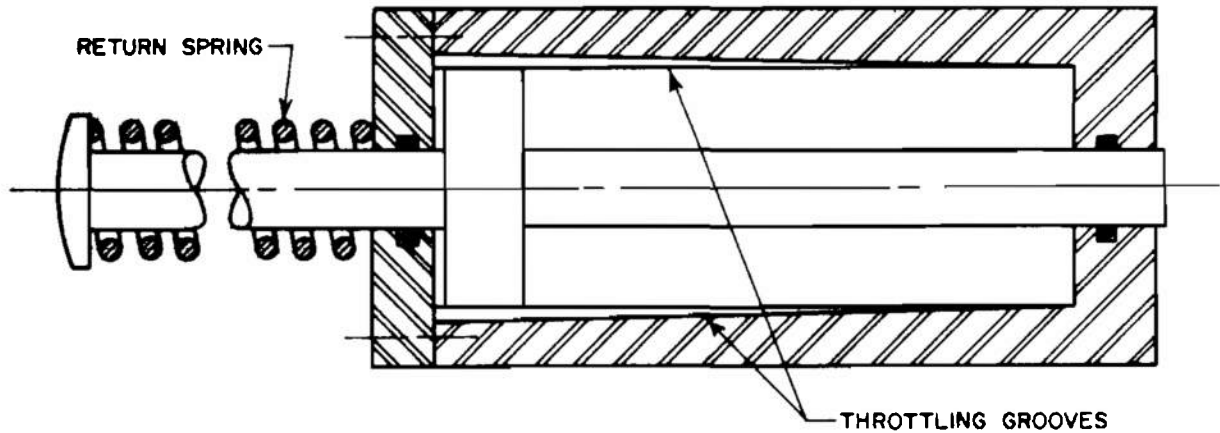‡ Reference 6, page 33, Equation 36.

19

Figure 20. Buffer

must have the same ratio, $v_x/\sqrt{F_b}$, for any given value of a_o. Under this circumstance, both v_x and F_b are unknown and can be found only by long, unwieldy, iterative computation. Since the velocity of the oil through the orifice controls the buffer force, any time that this velocity reaches design proportions for a given orifice area, the buffer force too must assume its corresponding design value. This may happen for a short distance along the buffer stroke if either or both traversing energy, E_t, and applied torque, T_R, are less than their design values.

58. Buffers are usually installed with their axes tangent to a circular arc passing through the initial point of contact. The radius, R_b, from the traversing axis to the point of contact should be large so as to decrease the force component induced by the applied torque, T_R. Although the point of contact moves outward during buffing, the line of action is always perpendicular to R_x, hence the moment arm of the resisting torque is always constant (see Figure 21). With force and moment arm being constant, the torque is also constant, hence the angular deceleration, α_b must be constant.

$$\alpha_b = \frac{\omega^2}{2\theta_{bm}} \quad (30)$$

where

ω = maximum angular velocity of traversing parts

θ_{bm} = maximum angular buffing distance

$$\theta_{bm} = \tan^{-1}\frac{S_b}{R_b} \quad (31)$$

The angular velocity at any position after the buffer is contacted is

$$\omega_b = (\omega^2 - 2\alpha_b\,\theta_b)^{1/2} \quad (32)$$

Then, according to the velocity diagram of Figure 21, the linear velocity of the buffer is

$$v_x = \omega_b R_x/\cos\theta_b \quad (33)$$

where

$$R_x = R_b/\cos\theta_b \quad (34)$$

The buffer stroke at any position θ_b is

$$x_b = R_b \tan\theta_b \quad (35)$$

V. SAMPLE PROBLEMS

PROBLEM A. TRAVERSING MECHANISM FOR STATIONARY WEAPON

59. Determine the characteristics for a traversing mechanism based on the following data for the traversing parts.

W_{tr} = 30,000 lb, weight
Φ = 10,000 slug-ft^2, mass moment of inertia
ω = 50°/sec, maximum angular velocity
α = 0.50 rad/sec^2, maximum angular acceleration
R_t = 5 in, radius traversing axis to CG
θ_t = 6°, slope of terrain
ϕ_a = 0°, location of CG with respect to horizontal line in plane parallel to the slope

From Equation 1

$M_w = W_{tr}R_t \sin\theta_t \cos\phi_a$
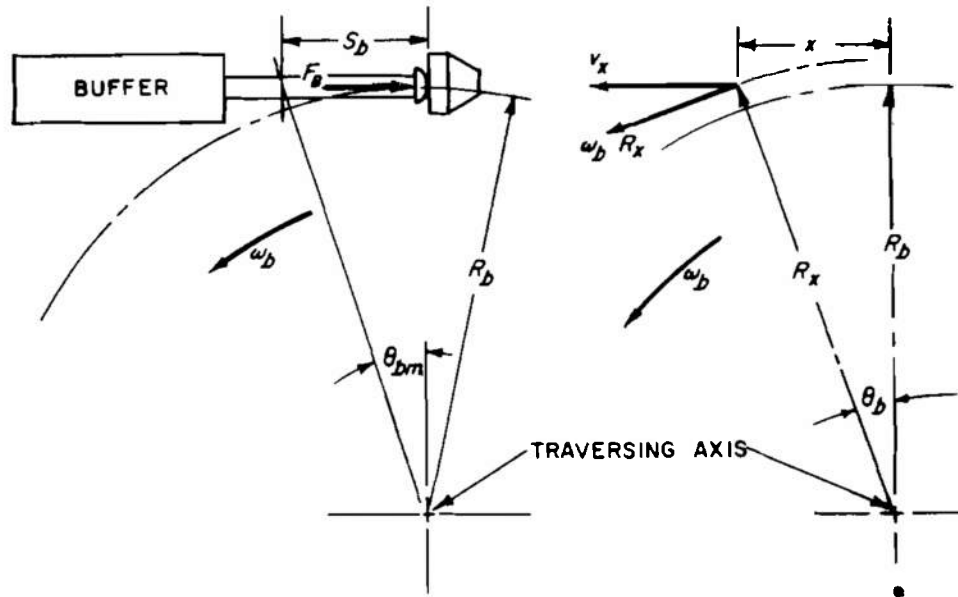
= 30,000 × 5 × .1045 × 1.0 = 15,700 lb-in

Figure 21. Diagram of Buffer Action

From Equation 2a

$$R_\mu = \frac{2}{3}\left(\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2}\right) = \frac{2}{3} \times \frac{9261 - 5832}{441 - 324}$$
$$= 19.55 \text{ in}$$

where

$$r_o = 21 \text{ in}; \ r_i = 18 \text{ in}$$

From Equation 2

$$T_b = \mu F_{br} R_\mu = 11{,}700 \text{ lb-in}$$

where

$F_{br} = 60{,}000$ lb, normal bearing load during firing (assumed)

$\mu = 0.01$, coefficient of friction

From Equation 3, for zero degrees elevation

$$T_f = (aF_g - bF_a) = 4000 \text{ lb-in}$$

where

$F_g = 339{,}000$ lb, propellant gas force
$F_a = 279{,}000$ lb, inertia force of recoiling parts
$a = 0.02$ in, distance, F_g to traverse axis
$b = 0.01$ in, distance, F_a to traverse axis

From Equation 4

$$T_a = \Phi\alpha = 10{,}000 \times 12 \times 0.5 = 60{,}000 \text{ lb-in}$$

From Equation 5

$$T_R = M_w + T_b + T_f + T_a = 91{,}400 \text{ lb-in}$$

This is the required torque at the traversing gear.

60. The gear ratio of the powered traversing mechanism is determined from the required traversing velocity and the rated speed of the motor which, in this case, is assumed to be 1800 rpm.

$$\omega = 50°/\text{sec} = \frac{50}{360} \times 60 = 8.33 \text{ rpm, traversing speed}$$

$\omega_m = 1800$ rpm, normal operating motor speed

$$r_g = \frac{\omega_m}{\omega} = 216, \text{ gear train ratio}$$

Selection of the gear train configuration is arbitrary although sizes of pinions and gears should be compatible. Figure 22 is a schematic illustration of the train whose gear ratio is

$$r_g = \frac{R_{P_8}}{R_{P_7}} \times \frac{R_{P_6}}{R_{P_5}} \times \frac{R_{P_4}}{R_{P_3}} \times \frac{R_{P_2}}{R_{P_1}}$$
$$= \frac{24}{3} \times \frac{5}{1.25} \times \frac{3}{1} \times \frac{2.25}{1} = 216$$

where R_{P_x} = pitch radius of the respective gears.

Table 3 is the accumulated data of the gear train. The procedure for obtaining the data is based on the material on spur gears in Reference 4. These are precision gears and

21

TABLE 3. *Gear Train Design Data, Stationary Mount*

Gear	R_{p_x} (in)	T (lb-in)	F_{xx} (lb)	ω (rpm)	V_p (fpm)	c_r
8	24	91400	3810	8.33	105	0.878
7	3	11540	3810	66.7	105	0.878
6	5	11540	2310	66.7	175	0.842
5	1.25	2910	2310	267	175	0.842
4	3	2910	970	267	419	0.756
3	1	980	970	800	419	0.756
2	2.25	980	435	800	943	0.634
1	1	440	435	1800	943	0.634

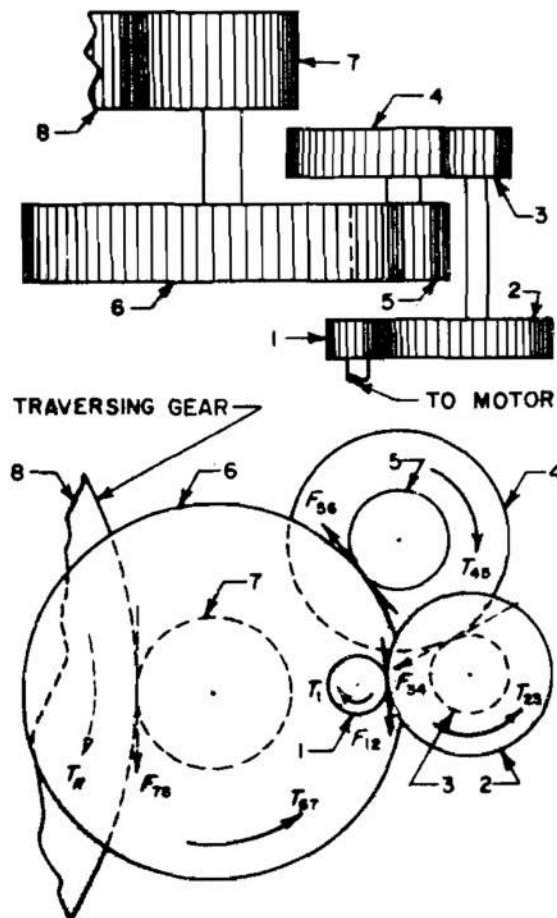| Gear	y'	N	P_r (in)	F_r (in)	C_i	F_w (in)
8	0.0008	336	0.448	1.344	0.64	$2\frac{1}{8}$
7	0.0008	42	0.448	1.344	0.64	$2\frac{1}{8}$
6	0.0029	80	0.393	1.179	0.85	$1\frac{7}{16}$
5	0.0029	20	0.393	1.179	0.85	$1\frac{7}{16}$
4	0.0021	72	0.262	0.786	0.90	$\frac{7}{8}$
3	0.0021	24	0.262	0.786	0.90	$\frac{7}{8}$
2	0.0012	72	0.196	0.588	0.90	$\frac{11}{16}$
1	0.0012	32	0.196	0.588	0.90	$\frac{11}{16}$

Figure 22. *Traversing Gear Train*

although a more elaborate method is available in the same reference, the one presented here is adequate for illustrating a gear design procedure. Since the teeth in pinions are weaker than the mating gear teeth, the calculations are based on pinion dimensions, No. 5 being selected for the analysis.

$$F_{56} = \frac{1}{\eta_{H}} \times \frac{T_{R}}{R_{p_8}} \times \frac{R_{p_7}}{R_{p_6}} = \frac{1}{0.99} \times \frac{91400}{24} \times \frac{3}{5}$$

$$= 2310 \text{ lb}$$

$$\omega_5 = \omega_6 \frac{R_{p_6}}{R_{p_5}} = 66.7 \times \frac{5}{1.25} = 267 \text{ rpm}$$

$$v_{p_5} = \frac{2\pi}{12} \times \omega_5 R_{p_5} = \frac{2\pi}{12} \times 267 \times 1.25$$

$$= 175 \text{ ft/min} = \frac{2\pi}{12} \times 8.33 \times 24 \times \frac{5}{3}$$

$$= 175 \text{ ft/min}$$

According to Equation (13–8) of Reference 4

$$y' = \frac{F_{56}}{\sigma_c c_v C_F D_p^2} = \frac{0.231}{79} = 0.0029$$

σ_c = 50,000 lb/in^2, endurance limit of gear teeth

C_F = 3, ratio, face width to circular pitch*

D_p = 2.5 in, pitch diameter of pinion No. 5

c_v = $1 - (\sqrt{v_{p_5}}/84)$ = 0.842, Barth's velocity factor‡

On Figure 13–17 of Reference 4, locate the intersection of the horizontal line corresponding to $y' = 0.0029$ and the curve for the 20° ASA stub tooth. The intersection falls on the

* Reference 4, page 334.
‡ Reference 4, page 333.

vertical line for $N_5 = 20$, the number of teeth on the pinion.

$$P_d = \frac{N_5}{D_p} = \frac{20}{2.5} = 8, \text{ diametral pitch}$$

$N_6 = D_{p_6} P_d = 10 \times 8 = 80$, number of teeth on gear No. 6

$$P_c = \frac{\pi}{P_d} = 0.393, \text{ circular pitch}$$

$$F_c = C_F P_c = 3 \times 0.393$$
$$= 1.179 \text{ in, effective face width}$$

From Table 13-5 of Reference 4, for a face width not exceeding 2 in, $C_i = 0.85$ for the second reduction gear. The actual face width is

$$F_w = \frac{F_c}{C_i} = \frac{1.179}{0.85} = 1.39, \text{ or } 1\frac{7}{16} \text{ in}$$

61. The torque at pinion No. 1, the drive motor pinion, is shown in Table 3 as $T = 440$ lb-in. The gears in the train are too small to add significantly to this torque. Later, calculation of a gear train composed of larger gears will show some difference in the horsepower required.

$HP = T\omega/33000$ when ω is in radians per minute

$$= \frac{T}{12} \frac{2\pi \text{ (rpm)}}{33000}$$

$$= \frac{440}{12} \times \frac{2\pi 1800}{33000} = 12.56$$

This is the maximum required horsepower at the motor shaft when the gear train is not included. The added effort to drive the gear train is almost negligible as demonstrated by the calculations below. The mass moments of inertia of the gears are listed in Table 4. Each gear is assumed to be a solid disk whose thickness is the face width and whose radius is the pitch radius. The assumption is conservative because the gears can be made lighter simply by removing material from the region between rim and hub.

TABLE 4. *Mass Moments of Inertia of Gears, Stationary Mount*

Gear x	R_{p_x} (in)	$R_{p_x}^4$ (in^4)	F_w (in)	Φ_x (lb-in-sec^2)	Φ_{xx} (lb-in-sec^2)
7	3	81	2.125	0.200	1.242
6	5	625	1.4375	1.042	
5	1.25	2.4	1.4375	0.040	0.122
4	3	81	.875	0.082	
3	1	1	.875	0.0011	0.0216
2	2.25	26	.6875	0.0205	
1	1	1	.6875	0.0008	0.0008

where

$$\Phi_x = \frac{1}{2} M R_{p_x}^2 = \frac{\delta\pi}{2g} R_{p_x}^4 F_w$$
$$= 0.00116 \, R_{p_x}^4 F_w \text{ lb-in-sec}^2$$
F_w = face width, in
R_{p_x} = pitch radius, in
g = 386.4 in/sec^2
δ = 0.285 lb/in^3, density of steel

From Equation 22

$$T_\alpha = \frac{\alpha}{\eta_g^{n-1}} \left(\Phi_{67} \frac{R_{p_1} R_{p_3} R_{p_5} R_{p_8}}{R_{p_2} R_{p_4} R_{p_6} R_{p_7}} \right.$$

$$+ \Phi_{45} \frac{R_{p_1} R_{p_3} R_{p_6} R_{p_8}}{R_{p_2} R_{p_4} R_{p_5} R_{p_7}} + \Phi_{23} \frac{R_{p_1} R_{p_4} R_{p_6} R_{p_8}}{R_{p_2} R_{p_3} R_{p_5} R_{p_7}}$$

$$+ \left. \Phi_1 \frac{R_{p_2} R_{p_4} R_{p_6} R_{p_8}}{R_{p_1} R_{p_3} R_{p_5} R_{p_7}} \right)$$

$$= \frac{0.5}{0.993} \left(1.242 \frac{1 \times 1 \times 1.25 \times 24}{2.25 \times 3 \times 5 \times 3} \right.$$

$$+ .122 \frac{1 \times 1 \times 5 \times 24}{2.25 \times 3 \times 1.25 \times 3}$$

$$+ 0.0217 \frac{1 \times 3 \times 5 \times 24}{2.25 \times 1 \times 1.25 \times 3}$$

$$+ \left. .008 \frac{2.25 \times 3 \times 5 \times 24}{1 \times 1 \times 1.25 \times 3} \right)$$

$$= \frac{0.5}{0.97} (0.37 + 0.58 + 0.93 + 0.17)$$

$$= 1.06 \text{ lb-in}$$

Since $T_i = T_1 = 440$ lb-in (see Table 3), the torque increase required for accelerating the gear train is less than 0.25 percent and therefore may be neglected.

PROBLEM B. TRAVERSING MECHANISM FOR MANEUVERING WEAPON

62. Determine the gear train characteristics for this weapon if, while traveling, it slews at an angular acceleration of $\alpha = 6$ rad/sec^2. The entire accelerating force to maintain the relative position between traversing parts and carriage is borne by the traversing mechanism. The motor is not required to develop this acceleration, it must merely hold against the reverse torque. Firing does not take place, therefore only the weight of the traversing parts rests on the traversing bearing and the frictional force here resists the induced torque. As in Paragraph 59

$$M_w = 15,700 \text{ lb-in}$$

From Equation 2

$$T_b = -\mu_b F_{br} R_\mu = -5,800 \text{ lb-in}$$

where

$F_{br} = W_{tr} = 30,000$ lb
$R_\mu = 19.55$ in, frictional radius
$\mu_b = .01$, coefficient of friction of bearing
$T_f = 0$ (no firing force)

From Equation 4

$$T_a = \Phi\alpha = 10,000 \times 12 \times 6.0$$
$$= 720,000 \text{ lb-in}$$

From Equation 9

$$T_R = M_w + T_b + T_a = 729,900 \text{ lb-in}$$

The gear ratio of the preceding problem is used, $r_g = 216$, and the gear design data entered in Table 5. It should be noted that this is a static condition, therefore $c_v = 1.0$. Also, the frictional losses sustained in the gear train help to reduce the reverse torque as it approaches the drive motor.

63. Detailed calculations are shown for pinion No. 5. The applied torque now being a reverse torque, the friction helps to resist it, otherwise the calculations are similar to those in Problem A.

$$F_{56} = \eta_g \times \frac{T_R}{R_{P_8}} \times \frac{R_{P_7}}{R_{P_6}}$$
$$= 0.99 \times \frac{729,900}{24} \times \frac{4}{10}$$
$$= 12,040 \text{ lb, tooth load}$$

$$y' = \frac{F_a}{\sigma_r c_r C_F D_p^2}$$
$$= \frac{12,040}{50,000 \times 1.0 \times 3.0 \times 25} = 0.0032$$

From Figure 13-17 of Reference 4,

$N_5 = 19$, number of teeth

$P_d = \dfrac{N}{D_p} = \dfrac{19}{5} = 3.8$ teeth/in, diametral pitch

$N_6 = D_{P_6} P_d = 20 \times 3.8 = 76$, number of teeth on gear No. 6

$P_c = \dfrac{\pi}{P_d} = 0.826$ in, circular pitch

$F_c = C_F P_c = 3 \times 0.826 = 2.478$ in, effective face width

From Table 13-5 of Reference 4, for a face width between 2.0 and 18.0 in

$C_i = 0.885 - 0.0175 \, F$, thus $C_i = 0.833$

$F_w = \dfrac{F_c}{C_i} = \dfrac{2.478}{0.833} = 2.95$ in, or 3 in face width

TABLE 5. *Gear Train Design Data, Maneuvering Mount*

Gear	R_{P_x} (in)	T (lb-in)	F_{xx} (lb)	ω (rpm)	V_p (fpm)	c_v
8	24	729900	30400	0	0	1.0
7	4	120400	30400	0	0	1.0
6	10	120400	12040	0	0	1.0
5	2.5	29800	12040	0	0	1.0
4	6	29800	4970	0	0	1.0
3	2	9840	4970	0	0	1.0
2	4.5	9840	2190	0	0	1.0
1	1.5	3250	2190	0	0	1.0

Gear	y'	N	P_c (in)	F_c (in)	C_i	F_w (in)
8	0.0032	114	1.325	3.975	0.574	7
7	0.0032	19	1.325	3.975	0.574	7
6	0.0032	76	0.826	2.478	0.833	3
5	0.0032	19	0.826	2.478	0.833	3
4	0.0021	75	0.502	1.506	0.90	$1\frac{3}{4}$
3	0.0021	25	0.502	1.506	0.90	$1\frac{3}{4}$
2	0.0016	87	0.325	0.675	0.92	$\frac{3}{4}$
1	0.0016	29	0.325	0.675	0.92	$\frac{3}{4}$

64. The drive motor is not required to perform against the reverse torque but it must

supply the additional power to operate the gear train. The additional effort in this case is not large but must be considered. The mass moments of inertia of the gears (Table 6) are obtained similarly to those of Table 4 (see Paragraph 61).

TABLE 6. *Mass Moment of Inertia of Gears, Maneuvering Mount*

Gear x	R_{p_x} (in)	$R_{p_x}^4$ (in⁴)	F_w (in)	Φ_x (lb-in-sec²)	Φ_{xy} (lb-in-sec²)
7	4	256	7	2.1	36.9
6	10	10000	3	34.8	
5	2.5	39	3	0.14	2.77
4	6	1296	1.75	2.63	
3	2	16	1.75	0.03	0.39
2	4.5	410	0.75	0.36	
1	1.5	5	0.75	0.004	0.004

where

$$\Phi_x = \frac{1}{2} M R_{p_x}^2 = \frac{\delta \pi}{2g} R_{p_x}^4 F_w$$

$$= 0.00116\, R_{p_x}^4 F_w, \text{ lb-in-sec}^2$$

F_w = face width, in
R_{p_x} = pitch radius, in
g = 386.4 in/sec²,
δ = 0.285 lb/in³, density of steel

From Equation 22

$$T_\alpha = \frac{\alpha}{\eta_0^{n-1}} \left(\Phi_{67} \frac{R_{p_1} R_{p_3} R_{p_5} R_{p_8}}{R_{p_2} R_{p_4} R_{p_6} R_{p_7}} \right.$$

$$+ \Phi_{45} \frac{R_{p_1} R_{p_3} R_{p_6} R_{p_8}}{R_{p_2} R_{p_4} R_{p_5} R_{p_7}} + \Phi_{23} \frac{R_{p_1} R_{p_4} R_{p_6} R_{p_8}}{R_{p_2} R_{p_3} R_{p_5} R_{p_7}}$$

$$\left. + \Phi_1 \frac{R_{p_2} R_{p_4} R_{p_6} R_{p_8}}{R_{p_1} R_{p_3} R_{p_5} R_{p_7}} \right)$$

$$= \frac{0.5}{0.99^3} \left(36.9 \frac{1.5 \times 2 \times 2.5 \times 24}{4.5 \times 6 \times 10 \times 4} \right.$$

$$+ 2.77 \frac{1.5 \times 2 \times 10 \times 24}{4.5 \times 6 \times 2.5 \times 4}$$

$$+ 0.39 \frac{1.5 \times 6 \times 10 \times 24}{4.5 \times 2 \times 2.5 \times 4}$$

$$\left. + 0.004 \frac{4.5 \times 6 \times 10 \times 24}{1.5 \times 2 \times 2.5 \times 4} \right)$$

$$= \frac{0.5}{0.97} (6.15 + 7.38 + 9.36 + 0.86)$$

$$= 12.2 \text{ lb-in}$$

From Table 3

$$T_t = T_1 = 440 \text{ lb-in}$$

The total output torque of the motor shaft according to Equation 23 is

$$T = T_t + T_\alpha = 452.2 \text{ lb-in}$$

The required horsepower output of the motor is

$HP = T\omega/33000$ when ω is in radians per minute

$$= \frac{T}{12} \frac{2\pi \, (\text{rpm})}{33000}$$

$$= \frac{452.2}{12} \times \frac{2\pi \, 1800}{33000} = 12.9$$

PROBLEM C. MANUAL TRAVERSING MECHANISM

65. Determine the characteristics of a gear train equipped with worm and worm gear for handwheel operation if data are the same as in the preceding problems. The schematic of the gear train is shown in Figure 23. Because handwheel operations are slow, effort is needed to overcome only static resistances.
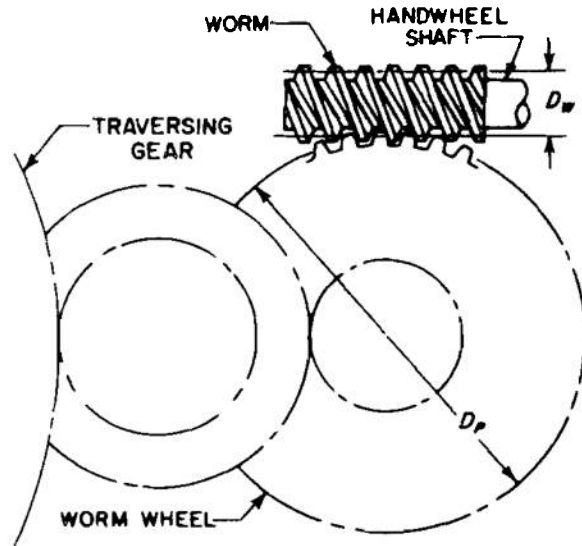

Figure 23. Gear Train, Manual Traverse

From Paragraph 62

$$M_{w} = 15,700 \text{ lb-in}$$
$$T_{b} = 5,800 \text{ lb-in}$$

and from Equation 7

$$T_{R} = M_{w} + T_{b} = 21,500 \text{ lb-in}$$

According to Paragraph 42, the gear ratio is set at 640:1. The required handwheel torque based on 100 per cent efficiency is

$$T_{t} = \frac{T_{R}}{r_{g}} = \frac{21,500}{640} = 33.6 \text{ lb-in}$$

The gear ratio of the worm and gear is 64:1, decreased from 640:1 in two steps as shown schematically in Figure 23. The worm is irreversible, locking against any reverse torque. The reverse torque is assumed to be 729,900 lb-in, the same as that calculated in Paragraph 62.

$$T_{R} = 729,900 \text{ lb-in}$$

$$T_{w} = \frac{T_{R}}{r_{g_{1}}} = 72,990 \text{ lb-in, torque on worm gear}$$

where

$r_{g_{1}} = 10$, the gear ratio between worm gear and traversing gear

66. The design data of the worm gearing is obtained by following the procedures in Article 15-6 of Reference 4. The worm is cast iron and the gear is phosphor bronze. As a trial measure, assume

P_{c} = 1.125 in, circular pitch of worm gear
P_{L} = 1.125 in, axial pitch of worm
N_{w} = 1.0, number of threads in worm
N = 64, number of teeth in worm wheel
β = $14\frac{1}{2}°$, pressure angle

Then,

$$P_{d} = \frac{\pi}{P_{c}} = 2.79 \text{ teeth/in, diametral pitch}$$

$$D_{p} = \frac{N}{P_{d}} = 22.9, \text{ pitch diameter of gear}$$

From Table 15-1 of Reference 4

$D_{w} = 2.4\ P_{L} + 1.1 = 2.7 + 1.1 = 3.8 \text{ in,}$
 pitch diameter of worm
$F_{w} = 2.38\ P_{c} + 0.25 = 2.68 + 0.25 = 2.93$
 or 3.0 in, face width

The lead angle, λ, is $5°23'$ because

$$\tan \lambda = \frac{P_{L}N_{w}}{\pi D_{w}} = \frac{1.25 \times 1.0}{3.8\pi} = 0.0942$$

According to Equations 15-17 and 15-18 of Reference 4, the limiting tooth load for wear is

$$F_{gw} = D_{p}F_{w}K_{w} = 22.9 \times 3.0 \times 150$$
$$= 10,300 \text{ lb}$$

and the limiting tooth load for beam strength is

$$F_{gs} = \sigma_{c}P_{c}F_{w}y$$
$$= 24,000 \times 1.125 \times 3.0 \times 0.1 = 8,100 \text{ lb}$$

The values of K_{w}, σ_{c} and y are obtained from Table 15-2 of Reference 4.

K_{w} = 150 lb/in², wear factor
σ_{c} = 24,000 lb/in², endurance limit in bending
y = 0.10, Lewis factor

The gear load, excluding frictional efforts is

$$F_{g} = \frac{T_{w}}{\frac{1}{2}D_{p}} = \frac{72,990}{11.45} = 6,370 \text{ lb}$$

Subsequent calculations for a larger and a smaller circular pitch show the following results:

For P_{c} = 1.0; F_{gw} = 8040 lb;
 F_{gs} = 6,310 lb; F_{g} = 7,170 lb
For P_{c} = 1.25; F_{gw} = 12,400 lb;
 F_{gs} = 9,750 lb; F_{g} = 5,720 lb

Neither of these is acceptable. The first has one of the allowable loads less than the actual tooth load. The second has too large a spread between the allowable loads and the actual loads.

67. The worm and gear are lubricated and protected from dirt to preserve a low coefficient of friction. However, low friction means a small lead angle if irreversibility is to be achieved. The coefficient of friction in this case is 0.10. Since $\tan \lambda = 0.094 < \mu = 0.10$, the worm cannot be driven by the gear and reverse torques will be locked out. With this required small lead angle, the efficiency of the worm gearing will be low.

From Equation 18

$$\eta_{w} = \frac{\cos \beta - \mu \tan \lambda}{\cos \beta + \mu \cot \lambda}$$
$$= \frac{0.968 - 0.10 \times 0.94}{0.968 + 0.10 \times 10.61} = 0.472$$

where

$\beta = 14\frac{1}{2}°$, pressure angle

If the friction in the thrust bearing of the worm is considered, then the efficiency according to Equation 18a is

where

D_b = 1.75 in, effective diameter of the thrust bearing

μ_b = 0.02, coefficient of friction of the thrust bearing

$$\eta_w = \frac{\cos \beta - \mu \tan \lambda}{\cos \beta \left(1 + \mu_b \dfrac{D_b}{D_w} \cot \lambda\right) + \mu \cot \lambda \left(1 - \mu_b \dfrac{D_b}{D_w} \tan \lambda\right)}$$

$$= \frac{0.968 - 10 \times 0.094}{0.968 \left(1 + 0.02 \times \dfrac{1.75}{3.8} \times 10.61\right) + 0.10 \times 10.61 \left(1 - 0.02 \times \dfrac{1.75}{3.8} \times 0.094\right)}$$

$$= \frac{0.959}{1.062 + 1.06} = 0.455$$

From Equation 17, the torque at the hand-wheel is

$$T_t = \frac{1}{\eta_w \eta_g^{n-1}} \frac{T_R}{r_g} = \frac{21,500}{0.455 \times 0.98 \times 640}$$
$$= 75.6 \text{ lb-in}$$

where

η_g = 0.99, efficiency of each spur gear mesh

n = 3, number of spur gear meshes

This required handwheel torque is higher than the recommended value of 50 lb-in. It must now be decided which is the more acceptable; a gear ratio of 640:1 which provides a convenient mil count at the handwheel (one revolution to 10 mils of traverse) or a larger gear ratio to provide an easier handwheel effort.

PROBLEM D. BRAKE

68. Determine the design data for an internal-shoe brake required to stop the traversing system of Problem B after it reaches the maximum rated angular velocity. The brake is installed on the shaft connecting gear 2 and pinion 3 of Figure 22. Here the velocity is 600 rpm which is equivalent to 1800 rpm at the motor shaft. The brake is designed for a stopping torque of $2\frac{1}{2}$ times the driving torque at gear unit 23. The procedures for computing the brake characteristics are those presented in Reference 4, Chapter 20. Figure 24 is a sketch indicating the dimensions and loads of the brake. The known data are

ω_{23} = 600 rpm, angular velocity of gears 2 and 3

T = 452.2 lb-in, output torque of motor (Paragraph 64)

D = 2R = 12 in, brake diameter at rubbing surfaces

w = 1.75 in, width of brake band

μ = 0.3, coefficient of friction

From Figure 24

a = 3.5 in; b = 5.0 in

ϕ = 10°; θ = 20°; β = 100°; $\psi = \beta + \theta = 120°$

Moments are taken about q, the pivots of the brake shoe, and consist of three components. These are computed according to Equations 20–11c, 20–12 and 20–13 of Reference 4 and apply to both left and right shoes. The sign
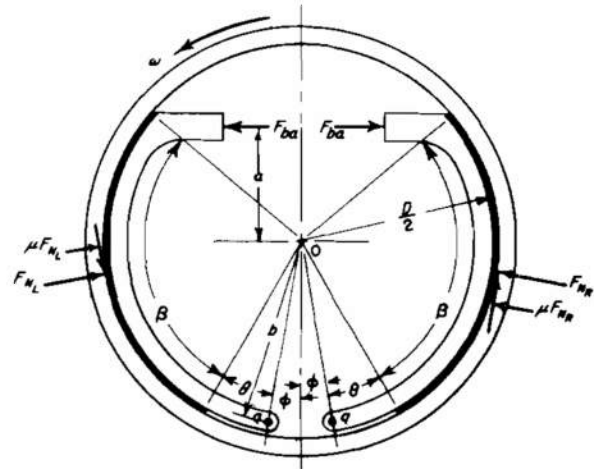

Figure 24. Brake With Applied Loads

of each component depends on the direction of rotation.

$$M_F = F_{ba} (a + b \cos \phi)$$
$$= (3.5 + 5 \times .985)F = 8.425 \, F_{ba}$$

$$M_N = \tfrac{1}{4} p_m \, bwD \, (\psi - \theta - \tfrac{1}{2} \sin 2\psi + \tfrac{1}{2} \sin 2\theta)$$

$$= \frac{1}{4} \times 5 \times 1.75$$

$$\times 12 \left(\frac{2\pi}{3} - \frac{\pi}{9} + \frac{1}{2} \times 0.866 + \frac{1}{2} \times 0.643 \right) p_m$$
$$= 65.6 \, p_m$$

$$M_f = \tfrac{1}{4} \mu \, p_m \, wD \, [D \, (\cos \theta - \cos \psi)$$
$$- \tfrac{1}{2} b \, (\cos 2\theta - \cos 2\psi)]$$

$$= \tfrac{1}{4} \times 0.3 \times 1.75 \times 12 \, [12 \, (0.940 + 0.5)$$
$$- \tfrac{1}{2} \times 5 \, (0.766 + 0.5)] \, p_m = 22.25 \, p_m$$

where p_m = maximum pressure on the brake band. According to Equation 20–14 of Reference 4, the braking torque for each shoe is

$$T_b = \tfrac{1}{4} p_m \, \mu D^2 \, w \, (\cos \theta - \cos \psi)$$
$$= \tfrac{1}{4} \times 0.3 \times 144 \times 1.75 \, (0.940 + 0.50) \, p_m$$
$$= 27.2 \, p_m$$

But

$$T_b = 2.5 \, T_{23}, \text{ according to specifications, and}$$

$$T_{23} = 2.5 \times \frac{R_{p_2}}{R_{p_1}} \, T = 2.5 \times 3 \times 452.2$$
$$= 3,390 \text{ lb-in}$$

The sum of the moments about the pivot of each shoe is equal to zero, therefore the total summation equals zero. The subscript, L, indicates the left brake shoe; R, the right brake shoe. If clockwise moments are positive, then

$$(M_N - M_f - M_F)_L$$
$$+ (-M_N - M_f + M_F)_R = 0$$

Since the brake shoes are symmetrical, $M_{F_L} = M_{F_R}$ and

$$M_{N_L} - M_{f_L} = M_{N_R} + M_{f_R}$$

Substituting the above values for M_N and M_f, we have

$$(65.6 - 22.25) \, p_{m_L} = (65.6 + 22.25) \, p_{m_R}$$

$$\frac{p_{m_L}}{p_{m_R}} = \frac{87.85}{43.35} = 2.025 \text{ or } p_{m_L} = 2.025 \, p_{m_R}$$

The total brake torque is

$$T_{b_L} + T_{b_R} = 3390 \text{ lb-in}$$

In terms of p_m

$$27.2 \, (p_{m_L} + p_{m_R}) = 3390 \text{ lb-in}$$

Substituting for p_{m_L}

$$3.025 \, p_{m_R} = 124.5 \text{ lb/in}^2$$

$$p_{m_R} = 41.2 \text{ lb/in}^2$$

$$p_{m_L} = 83.5 \text{ lb/in}^2$$

Balancing the moments about the pivots at q,

$$M_{F_L} = M_{N_L} - M_{f_L} = 43.35 \, p_{m_L} = 3620 \text{ lb-in}$$

$$M_{F_R} = M_{N_R} + M_{f_R} = 87.85 \, p_{m_R} = 3620 \text{ lb-in}$$

$$F_{ba} = \frac{3620}{8.425} = 430 \text{ lb, applied brake force}$$

69. The angular velocity of the brake is

$$\omega_{br} = \omega_{23} = 600 \, \frac{2\pi}{60} = 62.8 \text{ rad/sec}$$

The surface area of each brake shoe is

$$A_s = wR\beta = 1.75 \times 6.0 \times \frac{100}{180} \, \pi = 18.35 \text{ in}^2$$

Maximum torque on a shoe is

$$T_{b_L} = \frac{27.2}{12} \, p_{m_L} = 189 \text{ lb-ft}$$

The energy rate is

$$E_r = T_{b_L} \omega_b = 189 \times 62.8 = 11,870 \text{ ft-lb/sec}$$

The energy absorption rate is

$$E_a = \frac{E_r}{A_s} = \frac{11,870}{18.35} = 646 \text{ ft-lb/sec/in}^2$$

According to Article 20-4 of Reference 4, this value is acceptable.

The energy absorbed by the brake is

$$E_b = \tfrac{1}{2} \Phi_c \omega_{br}^2 = 47,400 \text{ lb-in}$$

where $\Phi_c = 24.05$ lb-in-sec^2, the effective mass moment of inertia at gear unit 23.

$$\Phi_c = \Phi \left(\frac{R_{p_3}}{R_{p_4}} \times \frac{R_{p_5}}{R_{p_6}} \times \frac{R_{p_7}}{R_{p_8}} \right)^2 + \Phi_{67} \left(\frac{R_{p_3}}{R_{p_4}} \times \frac{R_{p_5}}{R_{p_6}} \right)^2$$

$$+ \Phi_{45} \left(\frac{R_{p_3}}{R_{p_4}} \right)^2 + \Phi_{23} + \Phi_1 \left(\frac{R_{p_2}}{R_{p_1}} \right)^2$$

$$\Phi_c = \frac{120,000}{5200} + \frac{36.9}{144} + \frac{2.77}{9} + 0.39 + 0.004$$

$$\times 9 = 24.05 \text{ lb-in-sec}^2$$

$\Phi = 10,000$ lb-ft-sec^2 (see Paragraph 59)

For values of R_{p_x} and of Φ_{xx}, see Table 6.

$$\theta_{br} = \frac{E_b}{T_b} = \frac{47,400}{3390} = 14 \text{ rad, brake drum travel}$$

$$t_b = \frac{2\theta_b}{\omega_b} = \frac{28}{62.8} = 0.446 \text{ sec, braking time}$$

$$\alpha_{br} = \frac{\omega_{br}}{t_b} = \frac{62.8}{.446} = 141 \text{ rad/sec}^2, \text{ deceleration at brake}$$

$$\alpha_{tb} = \alpha_{br} \frac{1}{r_g} = \frac{141}{72} = 1.96 \text{ rad/sec}^2, \text{ deceleration of traversing parts due to braking}$$

PROBLEM E. SLIP CLUTCH

70. Compute the design data for a cone clutch which slips at a torque induced by 6 rad/sec^2 angular acceleration of the carriage. The clutch transmits the torque between gear 2 and pinion 3 shown in Figure 22. A sketch of the clutch, with loads, appears in Figure 25. The torque when slippage impends is $T_{23} = 9840$ lb-in (see Table 5). Other known data are

$R_2 = 4$ in, large radius of clutch at point of contact

$R_1 = 3.5$ in, small radius of clutch at point of contact

$w_c = 1.875$ in, width of clutch

$\phi = 14°55'$, slope of conical surface

$\mu = 0.25$, coefficient of friction

The radius of the friction circle is, according to Equation 2a,

$$R_\mu = \frac{2}{3} \left(\frac{R_2^3 - R_1^3}{R_2^2 - R_1^2} \right) = 3.76 \text{ in}$$

The torque to be transmitted by the clutch is

$$T_c = \mu F_N R_\mu$$

But

$$T_c = T_{23} = 9840 \text{ lb-in}$$

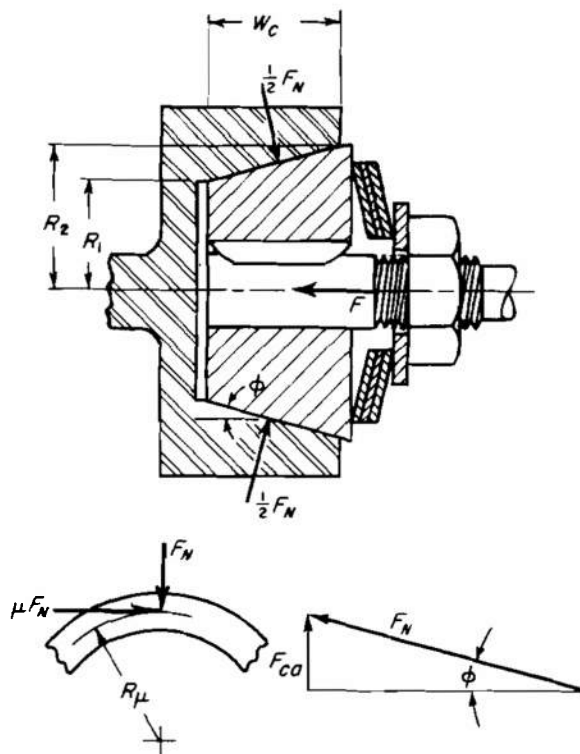The total force normal to the surface is

$$F_N = \frac{T_c}{\mu R_\mu} = \frac{9840}{0.25 \times 3.76} = 10,500 \text{ lb}$$

From the loading diagram of Figure 25, the thrust component of the normal force is

$$F_{ca} = F_N \sin \phi = 10,500 \times 0.2574 = 2700 \text{ lb}$$

The surface area of the clutch is

$$A_c = \frac{\pi}{\sin \phi} (R_2^2 - R_1^2) = 45.8 \text{ in}^2$$

PARTIAL END VIEW OF CLUTCH FACE   LOADING DIAGRAM (ROTATED 90°)

Figure 25. Clutch With Loading Diagram

The contact pressure of the clutch is

$$p_c = \frac{F_N}{A_c} = 229 \text{ lb/in}^2$$

71. The clutch should be completely enclosed for protection against dirt and weather so that variations in its frictional properties will be held to a minimum. It should be adjustable to the extent that variation in the coefficient of friction can be compensated for by variations in the axial thrust. The thrust is usually provided by a spring, either coil or Belleville, the latter being preferred because of its relatively small size and because large loads are available at small deflections. A typical installation is shown schematically in Figure 25. Calculations are shown below for the design data of a Belleville spring which will provide the desired axial thrust. The equation and procedure are found in Reference 7. Assume

$d_i = 1.5$ in, inside diam. of spring
$d_o = 3.5$ in, outside diam. of spring
$t = 0.05$ in, thickness of spring
$h = 0.25$ in, free height minus thickness
$\Delta = 0.025$ in, deflection of spring
$\nu = 0.30$, Poisson's ratio for steel

From curves on page 72 of Reference 8.

$$C_1 = 1.28, \ C_2 = 1.48, \ M = 0.74$$

The stress at a deflection of $\Delta = .025$ inch is

$$\sigma = \frac{E\Delta}{(1 - \nu^2) \, Ma^2} \left[C_1 \left(h - \frac{\Delta}{2} \right) + C_2 t \right]$$

where

$$a = \tfrac{1}{2} d_o$$

$$\sigma = \frac{29 \times 10^6 \times .025}{0.91 \times .74 \times 3.06} \lfloor 1.28 \, (0.25 - 0.0125) + 1.48 \times 0.05 \rfloor = 133,000 \text{ lb/in}^2$$

The load which will produce a deflection of 0.025-inch is

$$F_s = \frac{E\Delta}{(1 - \nu^2) \, Ma^2} \left[\left(h - \frac{\Delta}{2} \right) (h - \Delta) t + t^3 \right]$$

$$= \frac{29 \times 10^6 \times 0.025}{0.91 \times 0.74 \times 3.06} \, (0.2375 \times 0.225 \times 0.05 + 0.000125) = 984 \text{ lb/washer}$$

Three stacked washers will produce a spring load of 2952 lb. This force is near enough to the design force of $F = 2700$ lb. The correct load can be obtained by simply adjusting the deflection. Although greater deflections will increase the stress, the present stress of 133,000 lb/in² may reach 200,000 lb/in² before permanent set begins.

PROBLEM F. BUFFERS

72. Assume that the weapon in Problem A is one of limited traverse and buffers are needed to stop the traversing system. The buffers are located 32 inches from the traversing axis (R_b) and have a stroke of 5 inches (S_b). They are designed for the maximum loading condition, which means absorbing the rotational energy at maximum angular velocity while the torque of the drive motor is still applied. According to Equation 28, the constant buffer force is

$$F_b = \frac{E_t}{S_b} + \frac{T_R}{R_b} = 9200 + 2860 = 12,060 \text{ lb}$$

where

$E_t = \tfrac{1}{2} \Phi_e \, \omega^2 = 46,000$ in-lb, kinetic energy of the traversing system
$T_R = 91,400$ lb-in (see Paragraph 59)
$R_b = 32$ in, radius, buffer to traversing axis
$S_b = 5$ in, length of buffer stroke
$\omega = 50°/\text{sec} = .873$ rad/sec (see Paragraph 59)

The effective mass moment of inertia of each rotating part at the traversing gear is the actual moment of inertia times the square of the gear ratio to that part.

$$\Phi_e = \Phi + \Phi_{67} \left(\frac{R_{p_8}}{R_{p_7}} \right)^2 + \Phi_{45} \left(\frac{R_{p_8} R_{p_6}}{R_{p_7} R_{p_5}} \right)^2 + \Phi_{23} \left(\frac{R_{p_8} R_{p_6} R_{p_4}}{R_{p_7} R_{p_5} R_{p_3}} \right)^2 + \Phi_1 \, r_g^2$$

$$= 120,000 + 80 + 130 + 200 + 40$$
$$= 120,450 \text{ lb-in-sec}^2$$

where

$\Phi = 10,000$ lb-ft-sec² (see Paragraph 59)
$r_g = 216$ (see Paragraph 60)

The values of R_p are found in Table 3 and the values of Φ for the gears are found in Table 4.

73. From Equation 31, the maximum angular buffing distance is

$$\theta_{b_m} = \tan^{-1} \frac{S_b}{R_b} = \tan^{-1} \frac{5}{32}$$
$$= 8°53' = 0.155 \text{ rad}$$

The buffer stroke at any position θ_b, according to Equation 35 is

$$x_b = R_b \tan \theta_b = 32 \tan \theta_b$$

The constant angular deceleration is

$$\alpha_b = \frac{\omega^2}{2\theta_{b_m}} = \frac{0.762}{0.310} = 2.46 \text{ rad/sec}^2$$

(see Equation 30)

From Equation 32, the angular velocity at any position after the buffer is contacted is

$$\omega_b = (\omega^2 - 2\alpha_b\theta_b)^{1/2} = (0.762 - 4.92 \, \theta_b)^{1/2}$$

Converting to linear velocity according to Equation 33,
$$v_x = \omega_b R_r \cos \theta_b$$
where

$$R_r = R_b/\cos \theta_b = 32/\cos \theta_b \text{ (see Equation 34)}$$

74. Figure 20 is a schematic illustration of a buffer. The balanced piston offers the advantage of having a cylinder completely filled with oil regardless of the piston position. The spring returns the buffer to its in-battery position as the traversing parts are reversed. The spring force is small and ordinarily need not be included as part of the buffing forces. According to Equation 29, the required orifice area is

$$a_o = \frac{v_x}{c_o}\left(\frac{\rho A_b^3}{2F_b}\right)^{1/2} = 0.001136 \, v_x$$

where

c_o = 0.60, orifice coefficient
w = 0.0315 lb/in^3, density of oil
F_b = 12,060 lb, constant buffer force (see Paragraph 72)
$A_b = \frac{\pi}{4} (d_p^2 - d_r^2) = 5.16 \text{ in}^2$, piston area
d_p = 2.75 in, piston diameter
d_r = 1.0 in, rod diameter
ρ = w/g, mass density of oil

The data involved in computing the orifice area are listed in Table 7.

The buffer has two grooves cut lengthwise into the inner cylinder wall, 180 degrees opposed. Each groove is 0.375 in wide. The groove depth at each position is

$$d_g = \frac{1}{2} \frac{a_o}{.375} = 1.333 \, a_o$$

TABLE 7. *Buffer Data*

θ_b (rad)	cos θ_b	tan θ_b	R_r (in)	ω_b (rad/sec)	v_x (in/sec)	a_o (in^2)	x_b (in)
0	1.0000	0	32.00	0.873	27.95	0.0318	0
0.01745	0.9998	0.0175	32.00	0.822	26.30	0.0299	0.56
0.03491	0.9994	0.0349	32.02	0.769	24.62	0.0280	1.12
0.05236	0.9986	0.0524	32.04	0.711	22.78	0.0259	1.68
0.06981	0.9976	0.0699	32.08	0.647	20.81	0.0236	2.24
0.08727	0.9962	0.0875	32.12	0.578	18.64	0.0212	2.80
0.10036	0.9950	0.1007	32.16	0.518	16.74	0.0190	3.22
0.11345	0.9936	0.1139	32.21	0.452	14.65	0.0166	3.64
0.12653	0.9920	0.1272	32.26	0.374	12.16	0.0138	4.07
0.13963	0.9903	0.1405	32.31	0.274	8.94	0.0102	4.49
0.14836	0.9890	0.1494	32.36	0.176	5.76	0.0065	4.78
0.15505	0.9880	0.1563	32.39	0	0	0	5.00

TECHNICAL TERMS AND DEFINITIONS

aim. Point or direct a weapon so that its missile is expected to hit the target.

antibacklash device. Mechanism which applies static torque in two directions in a gear train so as to maintain tooth contact.

antibacklash device, axial spring type. Antibacklash device which utilizes spring loads acting along axes of helical gears.

antibacklash device, tangential spring type. Antibacklash device which utilizes spring loads applied tangentially to two adjacent, parallel gears having a common center.

azimuth. Direction expressed as a horizontal angle.

bottom carriage. Secondary supporting structure of a gun. It supports the top carriage.

buffer, traversing. Shock absorber which absorbs the kinetic energy of the traversing parts and brings them to rest.

cannon. Component of a gun, howitzer, or mortar consisting of the complete assembly of tube, breech mechanism, firing mechanism or base cap.

carriage, gun. Structure which transmits the forces resulting from firing of a weapon to the ground. In mobile weapons it also serves as part of the structure during transport.

clutch, slip. Clutch designed to slip at a predetermined torque, as distinguished from a positive clutch.

director. Electromechanical equipment capable of aiming the weapon in accordance with firing data fed to it.

double recoil gun. Weapon having two complete recoiling units.

elevating mechanism. Mechanism which elevates the tipping parts of a gun or launcher.

elevation. Angular position of the axis of a gun or launcher with reference to the horizontal measured in a vertical plane.

equilibrator, azimuth. Equilibrator which balances the traversing parts.

error signal. Signal in servomechanisms applied to the control circuit that indicates the misalignment between the controlling and controlled members.

fire control. Control over direction, volume and time of fire of weapons.

fire cycle. Sequence of operation of a weapon from loading through firing.

firing couple, traverse. Horizontal couple about the traversing axis generated by the firing forces.

firing jack. Adjustable device which levels and supports a weapon during firing.

firing platform. Structure resting on the ground which provides a means to rotate a weapon for general positioning purposes.

friction circle. Circle on which a distributed frictional resistance to rotation may be considered to be concentrated.

gear ratio. Ratio of input speed to output speed in a gear train; the mechanical advantage of gears or gear train.

gear train. A group of meshing gears operating in sequence to transmit motion or power.

handwheel. Wheel designed for readily applying manual effort to operate a gear train.

hydraulic motor. Motor operated by the pressure exerted by a liquid and derived from a pump in a closed cycle.

in-battery. Normal position of all components of a weapon in condition to be fired.

launcher. Device for launching or assisting the launching of aircraft rockets, missiles, or similar apparatus.

laying. Directing or adjusting the aim of a weapon.

locking device. Device which prevents accidental or undesired motion of another device.

missile. Object that is, or is intended to be, thrown, dropped, projected, or propelled for the purpose of striking a target.

mount, gun. Supporting structure of a gun.

mount, single trail. Gun mount which is provided with one rearward thrust member or trail.

mount, split trail. Gun mount in which the trail is composed of two rigid members hinged at the carriage to be brought together for transport and spread apart for firing.

no-bak device. Provision in the traversing gear train to permit free motion in either

direction from the power source and prevent motion from a reverse torque.

off-target position. Any position of the components of a weapon other than the on-target position. Cf: **on-target position.**

on-target position. Required position of all components of a weapon to score a hit.

pitch. Angular displacement about a lateral axis.

position coordinates. Aiming conditions as defined by angular traverse and elevation positions.

power drive. Power source of an electric or mechanical driven mechanism.

projectile. Missile for use in any type of gun.

propellant gas force. Force exerted by the propellant gases on the breech over an area equal to the bore area.

propellant gas period. Duration of the propellant gas activity.

pump, constant displacement. Pump that has a constant stroke.

pump, variable displacement. Pump whose flow varies with length of stroke.

range, traverse. Angular distance through which a weapon can be moved by the traversing mechanism.

recoil force. Total resistance to movement of the recoiling parts.

recoiling parts. Unit consisting of all components of a weapon which move in recoil.

recoil mechanism. Unit which absorbs the energy of recoil.

self-propelled weapon. Weapon permanently installed on a vehicle which provides motive power for the weapon.

single recoil gun. Weapon having only one recoil system.

slew. Pivoting of ground vehicles while maneuvering.

small arms. Guns whose bore diameters do not exceed 30 millimeters (1.181 inches).

stabilizer. Unit which tends to maintain the on-target position of the weapon as a tank maneuvers.

tank. Self-propelled, heavily armored offensive vehicle, with fully enclosed revolving turret and equipped with at least one major weapon.

target. Object or area to be hit or at which the weapon is aimed.

top carriage. Upper structure of a gun carriage.

torque, reverse. Torque induced by the maneuvering accelerations of a vehicle.

tracking. Keeping a weapon aimed at a moving target.

trail. Rearward thrust member of a weapon. It stabilizes the weapon during firing and serves as a connecting link between weapon and prime mover during transport.

traverse. Horizontal angular movement of a weapon in either direction.

traverse, coarse. General positioning of a weapon in azimuth.

traverse, fine. Precise positioning of a weapon in azimuth.

traversing axis. Axis about which the weapon rotates to change direction in azimuth.

traversing bearing. Bearing on which the traversing parts rotate.

traversing gear. Large gear or gear segment which is the final member of the traversing gear train.

traversing mechanism. Mechanism by which a weapon can be rotated in a given plane.

traversing mechanism, axle. Mechanism which changes the direction of fire in azimuth by moving the cannon along the axle of the carriage.

traversing mechanism, ball joint. Mechanism which traverses the weapon by turning it about a pivot or ball joint.

traversing mechanism, base ring and racer. Mechanism which traverses the weapon by turning it on a bearing of large diameter. The base ring is fixed by the emplacement; the racer bears on the base ring.

traversing mechanism, pintle. Mechanism in which a weapon rotates on relatively small bearings about a short stanchion or pintle.

traversing parts. Unit consisting of all components of a weapon that move in traverse.

traversing system. Complete group of parts, moving and fixed, associated with traverse activity.

weight moment, traversing. Moment about the traversing axis caused by a non-level condition of the weapon.

REFERENCES

1. *Hand Crank Performance for Various Crank Radii and Torque Load Combination*, U. S. Army Ordnance Technical Memorandum No. 3–57, Human Engineering Laboratory, Aberdeen Proving Ground, Maryland.

2. ORDP 20–343, Ordnance Engineering Design Handbook, Carriages and Mounts Series, *Top Carriages*, September 1960.

3. M. F. Potts, *Design of Machine Elements*, Prentice-Hall, Inc., New York, 1957.

4. S. J. Berard, E. O. Walters and C. W. Phelps, *Principles of Machine Design*, Ronald Press Company, New York, 1955.

5. *Hydraulic Drives and Mechanical Systems*, Vol. 1, Section 5.0, "Evaluation of Hydraulic Components," The Franklin Institute, Interim Report I-2124-5, Contract No. NOrd 10625 BuOrd.

6. ORDP 20–342, Ordnance Engineering Design Handbook, Carriages and Mounts Series, *Recoil Systems*, February 1960.

7. *Handbook of Mechanical Springs*, Associated Spring Corporation, Bristol, Conn., 1958.

INDEX